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A. S. R. E. Journal

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July, 1915, to June, 1916

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PUBLISHED BI-MONTHLY

BY

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

154 NASSAU STREET, NEW YORK, N. Y.

Subscription price, United States and dependencies, $3.00 a year; Foreign countries in Postal Union, $4.00 a year.

Application for entry as second-class matter at the Post Office at New York, N. Y., pending.

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THE DESIGN, CONSTRUCTION AND OPERATION OF A THOUSAND TON AMMONIA COMPRESSION REFRIGERATING MACHINE*

By F. L. Fairbanks, Boston, Mass.

The Quincy Market Cold Storage & Warehouse Company, Boston, Mass., some time ago added four new cold storage warehouses and was obliged to consider the installation of an addition to its power-house equipment to take care of the increased load. As it was impossible to extend the Sargents Wharf power house, for the reason that additional land could not be purchased for the purpose, it was necessary that the added refrigerating equipment should be so concentrated as to be installed in such vacant space as might be available in the existing power plant.

The daily load in sight figured approximately 800 tons of refrigeration and, to take care of emergencies and a reasonable increase, it was decided that a capacity of 1,000 tons was necessary, and, owing to the limited space available, this must be in one unit.

The compression system, both single and double-acting, was considered, as well as the absorption system and, for various reasons, including local conditions, it was decided to adhere to the single-acting vertical type of compression machine, directly connected to a horizontal, cross-compound steam engine, this being the type of machines already installed in this plant.

Under the new conditions this station would be obliged to handle two large and entirely distinct brine systems; first, a warehouse system, which would ultimately require four or five million gallons of —12 degrees brine daily and, secondly, a street system, which would require six to ten million gallons of +10 degrees brine daily. On account of the size and constancy of operation of the two systems, it was possible to undertake to combine in some way the suction gases, so that they might be handled by a single compressor, and discharged into a single main condensing system.

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* Read at Tenth Annual Meeting, New York, N. Y., November 30 and December 1, 1914.

After very careful consideration it was decided to design a double-effect, three-cylinder compressor, which, while handling both systems at the same time, would have some compensating device by means of which variations in the relative load could be positively taken care of, so that within reasonable fluctuations the speed of the machine could be kept at a minimum, with a maximum compressor efficiency.

The capacity of the refrigerating machine, as well as the steam horsepower required, was figured on the basis of the suction gas pressure from the street brine-cooling system, which averages about 20 pounds, gauge, with a reasonable allowance for emergencies, such as a drop in the steam pressure, excessive head pressure and such other abnormal conditions as obtain in practical operation.

It was necessary that this equipment be ready for service before the very warm weather of the approaching summer set in and, inasmuch as there were but a little over eight months available in which to accomplish this work, steps were taken immediately to see if such a unit could be constructed by some compressor builder in the limited time allowed for the work of construction and erection.

After an investigation of the facilities of the various shops throughout the country it was found that there was no builder in a position to handle the work in the specified time, most of them not being in a position to handle the work at all on account of the size of the machine and rigidity of the specifications. It was then decided to design and erect this machine with our own men, contracting for the inidividual parts with several of the largest machine, foundry and forge shops, so that all work could be carried on simultaneously and, in so far as possible, have the machine work done by the same concerns that made the castings, thus saving considerable time and cost of transportation should a casting prove defective. Finally, the main contract was given to the Providence Engineering Works, Providence, R. I., a large firm of engine builders of known ability, and the various details were sublet to those concerns that were in the best position to handle them.

The contracts were based upon a price per pound for each detail, machined as per drawing, and based upon the estimated weights of the designer, with a provision that should these weights be high or low the contractor would be paid on the basis of actual weights of castings as delivered to shop to be machined. Where special prices for difficult machine work were submitted, these prices should remain fixed regardless of weight.

The Quincy Market Company furnished drawings, patterns, foundation, cranes, rigging and men for the erection and agreed to pay 85 per cent each month of the cost of all material and labor expended by the contractor. The responsibility for every piece of work for which the contractor could show a drawing approved by the designer rested with the Quincy Market Company, provided the work was to drawing and of sound material.

The drawings carried the specifications for each different material and, where necessary, physical and chemical characteristics. All important measurements, standing and running fits, clearances for expansion, as well as running clearances, were shown on the drawings in thousandths of an inch, thus making it possible to have multiple parts machined by different men and in separate shops without reference to each other, and at the same time have those various parts fit together accurately and as intended when assembled at the power plant of the Quincy Market Company. This worked out splendidly and proved a great time-saver.

Drawings, pattern work and castings were carried forward simultaneously, the blueprints for the low-pressure bed-plate being in the hands of the foundry in three weeks from the time work was started, and the fifty-ton casting was in the machine shop in ten weeks. Drawings, patterns and castings followed each other consecutively and in the order in which it was desired to have them reach the plant for erection, and without waiting for assembly drawings or anticipating the completed machine, except in so far as the designer considered it necessary for safety.

The designer was to check, test, and accept or reject each individual piece and reserved the privilege of observing or supervising any part of the work when he deemed it necessary.

The patterns were checked before and after moulding, the moulds were checked after cores were in and before closing, and castings were carefully examined to detect defects before being machined, all with a view to find and rectify any errors as early as possible, before such errors had developed into time and money. This proved an efficient method and, while it kept the designer on the jump, traveling nights and working twelve to sixteen hours a day, the results justified the method, as there was no part of this machine rejected for defect, either in material or workmanship, after it arrived at the power plant for erection.

The bed plates, A frames and flywheels were cast and machined by the Farrell Foundry & Machine Company, Ansonia, Conn.; the crankshaft, connecting rods and large forgings by the Bethlehem Steel Company at Bethlehem, Pa.; cranks, crossheads, piston spiders and other large steel castings by the Chester Steel Casting Company, Chester, Pa. Bronze castings, of which there were nearly fifteen tons, came from Bridgeport, Conn., and Philadelphia, Pa. The steam and ammonia cylinders, valve gear, governing mechanism and other engine and compressor details were machined in the plant of the Providence Engineering Works and cast or forged in Providence, Boston and surrounding places.

All finished work to be delivered f.o.b. cars on the Quincy Market Company's railroad siding at its Sargents Wharf power house. At this point the Quincy Market Company assumed all responsibility for handling and erecting on foundations as required, with its own men and rigging, this being deemed advisable owing to the fact that practically all material had to be carried on cranes over running engines. This being attended naturally with a great deal of risk, especially as the fifty-ton crane, with all the heavy rigging, was new and untried and the failure of a chain or other part at the time of passing over one of the other large compressors in service would not only wreck them, but would in all probability put the whole plant out of commission and possibly cause loss of life through the breaking of steam and ammonia mains. At these times a man was stationed at every important steam and ammonia valve with orders to shut it if he heard a crash. All rigging was so carefully tested that the entire work was accomplished without accident of this nature.

During erection the contractor was to furnish skilled shop labor for assembling, drilling and bolting together and such other work as would be done if erected in the shop. The power plant already having electric and pneumatic tools for repair work, the Quincy Market Company allowed the contractor the use of all such equipment in assembling, this making the work as convenient at the plant as it would have been in the shop.

The erection was facilitated by arranging for the shipment of the bed plates as soon as machined. These were immediately placed upon their foundation, grouted by ramming, hauled down by the foundation bolts and, after allowing sufficient time for hardening of the cement, the seats for the main bearing shells were bored in position by means of a large cylinder boring bar. Upon these bed plates each part was placed in position as fast as it was received from the various firms. The whole machine was turned over and put in service in a little over eight months from the beginning of the work.

There are three 30 x 50-inch ammonia cylinders on vertical A frames, driven by a 32 x 64 x 50-inch horizontal, cross-compound Corliss engine. The steam cylinders are steam jacketed, including the heads. The high-pressure cylinder uses steam at 150 lbs. pressure per square inch, superheated 100 degrees F., the low-pressure cylinder receives steam at 20 lbs. pressure from the high-pressure exhaust, which is superheated 50 degrees by passing through a reheating receiver, all jacket condensation being discharged through traps into the reheating receiver and, after being re-evaporated and superheated, becomes a part of the supply to the low-pressure cylinder, which exhausts into a vacuum of between 27 and 28 inches.

The ammonia cylinders have two suction connections through which the gases from a street pipe line and a warehouse system are taken from shell brine coolers. The gas from the warehouse system enters the suction valves at a pressure of from 4 to 5 lbs., gauge, and, after following the piston for 48 inches of the stroke, it is compressed to approximately one-half its volume by the admission, through piston ports, of gas from the street pipe system at a pressure of 20 to 21 lbs., gauge. The total volume at this pressure is discharged into an insulated gas header, from which the gas flows to shell type condensers through a feed-water heater, where their temperature is reduced to the saturation point. From the condensers the ammonia flows to a liquid receiver and then through a gas trap to the automatic cooler feeder.

The boiler feed water is taken from the city mains, passes through the ammonia discharge gas heater, and from there through an exhaust steam heater, located between the engine exhaust and the steam condenser, then through an economizer and finally reaching the boiler at a temperature of from 275 to 320 degrees F., while the flue gases leave the economizer at 175 to 212 degrees F.

Each compressor valve is in a separate cage, and can be removed and replaced with a new one in from three to five minutes by the removal of a screwed plug, which holds the valve cage in position.

The ammonia cylinders are water jacketed on the upper 26 inches of the stroke only, the water inlets being so arranged as to stratify the water as much as possible, the overflow being at the top. The low-pressure suction connections to each cylinder are 8-inch, with a 12-inch header, and 12-inch, with a 12-inch header, on the high-pressure suction. The discharge connections are 8-inch and the compressed gas passes through an automatic non-return valve to a 12-inch header on its way to the condensers.

Stop valves are provided at all important points and a four-inch safety by-pass, connected in parallel with a four-inch positive by-pass, is located between the suction and discharge headers and a positive equalizing valve is provided between the two suction headers to regulate the relative work done on either system, so that constant temperatures to within a degree can be obtained, without favoring one system to the detriment of the other and maintain, at the same time, the maximum efficiency of speed regulation.

Nine twelve-inch mercury thermometers, reading to one-quarter of a degree Fahr., are attached to permanent wells in each suction and each discharge connection to the compressor cylinders, giving at all times a positive indication as to the condition of the gases, as well as giving what is probably the best practical method of determining the working condition of piston, valves, etc. There is also a like thermometer on each water jacket cooling water supply and overflow.

All gauges are combined recording and indicating, the speed is indicated by an electric tachometer and a revolution counter, and the position of the cranks is shown by a crank indicator, all of which are mounted on pedestal gauge boards set near the valve stanchions and permanently connected.

The engine governor is of the bell-crank type, entirely enclosed, and is driven from the main shaft by cut steel gears and shafting through a selective gear transmission with six gear changes. A notched segment plate with pendulum weight accomplishes speed changes up to 12 revolutions. This, in connection with gear shifts in the transmission, allows a speed variation from 15 to 75 r. p. m. in increments of one-third of a revolution.

All steam used in the engine jackets, cylinders and re-heater is shut off by the closing of the throttle, in connection with which is an automatic double-ported engine-stop held open unbalanced, and closed instantly by steam through a solenoid valve operated electrically by either a fly-ball speed limit, broken governor gear or push-button.

All dash-pot valves can be removed and replaced while engine is in operation. This is also true of the governor spindle thrust bearing.

All oil drains to the crank pits and from the pits to a water washing device, and from there to a primary filtering and settling tank, from which it is pumped to French flannel filtering and storage tanks. Then it flows by gravity through an oil cooler to the machine.

The oil is kept at a temperature of 75 degrees F.; the thermometer on the oil line is read every hour and noted on the log with the other readings. A sample of the oil is tested for gravity and viscosity every morning.

The high-pressure steam connection is 10-inch and the low-pressure exhaust connection is 20-inch, leading into a 24-inch main exhaust header, connecting with all steam condensers.

All steam and ammonia valves and piping are identical and of the special design of the Quincy Market Company. They are of the globe pattern, no gate valves being used and no threaded or extra heavy pipe is used in sizes over 2½-inch. No shims or gaskets were used in the construction of the engine. The ammonia line gaskets are of special lead antimony placed in flat-face joints, and all steam lines have corrugated English manganese bronze gaskets in flat-face joints, the object being to make all steam and ammonia valves, pipe and fittings interchangeable, with the exception of the gaskets.

The steam engine, which was designed for economy, stability and flexibility of operation, has triple-ported Corliss valves requiring a minimum of movement for a maximum opening and separate eccentrics for driving each admission and exhaust gear. Each admission valve is independent of the other, so that wear or minor accident to one valve can be repaired without interfering with the other and without stopping the machine. The idea being, in so far as practicable, to design a machine which could be run twenty-four hours a day, month after month, under severe conditions of service, stand up to its work with a reasonable amount of care and allow of the minor repairs and adjustments necessary without shutting down.

Major repairs made necessary by carelessness or accident are provided for by building with the machine, to micrometer measurements, such extra parts as practice has shown are liable to be needed, having them ready for use and by having all rigging necessary to strip any part of the machine readily at all times. Two 100-ton jacks are permanently set in the flywheel pits, with top saddle castings fitting the rim of the 70,000-pound flywheels, so that by removing the main bearing cap bolts the main shaft, flywheels, connecting rods, crossheads and pistons, weighing over 150 tons, can be raised and new bearing shells inserted without delay and without disturbing a bolt or nut other than those which hold down the main caps.

The steam end of the plant has produced an indicated horsepower with 10½ pounds of steam and 1.1 pounds of coal.

The compressor cylinders, with a piston clearance of 5/1000 of an inch, have shown an average efficiency of 90 per cent and the friction loss is approximately 6 per cent. The average of a large number of observations has shown that 94 per cent of the energy of the steam engine is delivered to the ammonia cylinders. These efficiencies are due mainly to modern design and construction, high-grade material, excellent workmanship and favorable local conditions.

The idea of compressing the suction gases from two different systems and at two widely different pressures, under the conditions which obtain at this plant, has been thoroughly worked out in practice during some two years of constant operation, and its efficiency, in the opinion of the writer, under these conditions, is limited principally to the saving accomplished in combining in one large machine, with its higher steam economy, the refrigerating load of two separate systems, which would otherwise be handled by two separate smaller machines, with the consequent increase in cost of oil, labor and attention required, as well as probable decreased steam economy of the smaller machines, especially if neither of the latter was carrying the load best suited to its efficient operation.

The maximum capacity of the ammonia cylinders under these conditions is, of course, that due to the higher or street system gas pressure and is constant regardless of fluctuations in pressure of the low-pressure gas.

An advantage in very low temperature work, with suction pressure below atmosphere, is that the ammonia rod packing box is always subject to the higher gas pressure, so that the danger of air admission at this point is eliminated.

The suction valve design must be given careful consideration, as this feature alone, unless very carefully worked out, will nullify practically all of the benefits to be derived. The admission of the high-pressure gas through the piston ports is in the nature of an explosion, and this at a time when the suction valves will be wide open, the result being a very forcible closing of these valves. Large area, low lift, good lubrication and quick action being absolutely necessary to prevent excessive hammering at the point of admission through the ports and, with all these combined in a design worked out for known local conditions, smoothness of operation begins to disappear at between 35 and 40 revolutions per minute. Speeds above 50 revolutions per minute are attended with more or less valve slamming, rapidly increasing discharge temperatures and an increase in horsepower per ton.

While 75 revolutions per minute is readily obtained with this compressor without valve distress when operating with the high- or low-pressure gas individually, operation is practically limited to 50 revolutions per minute when operating combined.

Figures 1, 2 and 3 are reproductions of diagrams from the ammonia end of the compressors. Figure 1 shows the compressor cylinders operating with low-pressure gas from the warehouse system. Figure 2 shows the same with gas from the high-pressure or street system, and Figure 3 shows the machine operating with the combined gases.

A large number of indicator cards show that the suction line on the indicator cards is practically identical with the same pressure line taken from the suction header. This probably is due to the fact that the suction valve area is nearly seven per cent of the piston area and to the suction connections being short and excessively large.

It will be noted on the indicator card taken when the gases were combined that the piston ports are also large enough to allow of the filling of the cylinder at speeds up to 50 revolutions per minute before the piston has reached the end of its stroke.

To the very small clearance is probably due the fact that there is practically no re-expansion of gases, it being possible to produce practically square corners on a card by feeding a few extra drops of oil per minute to the cylinders.

Figure 3 shows very plainly that the piston ports are recovered before compression can force the gas back through the ports and the entire valve action in this compressor, as shown by the many indicator cards taken, is so exceptionally good that it considerably exceeds the expectations of the designer, especially when operating with a single gas only and at high speed.

The horsepower required per ton of refrigeration with the combined gases is from five to ten per cent greater at low speeds than with the high-pressure gas alone producing the same tonnage, and at the higher speeds, say from 40 to 60 revolutions per minute, the horsepower required is from ten to twenty per cent greater. The reasons for these discrepancies have been demonstrated thoroughly by two years' operation, but it is hardly possible to go into sufficient detail in this paper. In the writer's opinion, it is doubtful if this feature is advisable in small machines unless the local con-

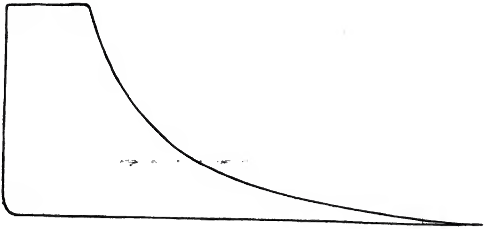

FIG. 1.—Condenser pressure, 123 lbs.; suction pressure, atmospheric; scale of spring, 80 lbs.; speed, 30 r. p. m.; cylinder, 30" x 50".

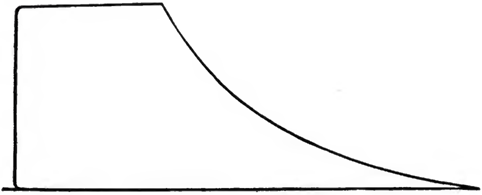

FIG. 2.—Condenser pressure, 123 lbs.; suction pressure, 17 lbs.; scale of spring, 80 lbs.; speed, 30 r. p. m.; cylinder, 30" x 50".

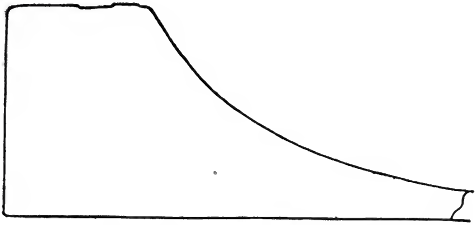

FIG. 3.—Combination of Figs. 1 and 2, showing the effect of the introduction of the higher suction pressure.

ditions are such that a considerable saving can be made in the operation, other than that which is due to any increase in compressor capacity alone.

The ammonia condensers are of the shell type, designed by the writer, with a rated capacity of 75 tons each and capable of easily carrying 100 tons when necessary. Water from the harbor is used for condensing and is first pumped through the ammonia condensers, after which it flows by gravity to the surface steam condensers and then back into the harbor. In operation from six to ten degrees in temperature is added to the water at the ammonia condensers and from fifteen to twenty degrees in the steam condensers, depending upon the temperature of the harbor water and the refrigerating load carried.

With thermometers located in the overflow lines the operating men can check the work of these condensers very closely by means of the inlet and outlet temperatures, and thereby get an accurate indication of the amount of water necessary with any load. As all of the condensing water is pumped through 12-inch Venturi meters and constantly indicated on the manometers, not only the amount pumped but also the actual work being done is indicated to the operator at all times, as well as being logged for record every hour.

These condensers are vertical and the gas from the compressor enters the shell tangentially at the top, flowing downward by gravity. The condensing water entering the bottom and overflowing at the top through a side outlet, so that the top ends of the tubes are open for inspection and cleaning at all times without interfering with the operation of the condenser. This cleaning is done by compressed air, or with a tube cleaning brush, but the compressed-air method is used almost entirely.

These condensers have 48-inch forged-welded shells, contain 367 one-and-a-half-inch 10-gauge charcoal iron tubes, and have no ammonia joints in the water except the tube ends, the lower ends always being sealed with oil coming from the compressors. The upper ends being visible at all times, the slightest leak is easily detected.

A U expansion joint is rolled in these shells which relieves expansion and contraction stresses, and this, together with the method of rolling the tubes into the tube sheet, has almost entirely eliminated the trouble from leakage, which has usually been experienced with shell condensers.

An added feature of these condensers, in addition to the facility with which they may be cleaned and inspected, is that when pinhole leaks develop in a tube, due to a defective spot in the metal, or salt water corrosion, the leak is not only quickly detected but the entire charge in this condenser can be blown into a still, the old tube removed, a new one rolled in and the condenser put in service again in from an hour and a half to two hours.

The still, which is of the same general construction as the condensers, has connected to it a header and to which each of the condensers is permanently connected, so that the charge of any condenser, by simply opening and closing the proper valves, may be thoroughly cleaned and pumped back by a small compressor into the main system through a caustic soda rectifier, thereby making it possible to purge and cleanse the ammonia in the system quickly and practically without loss at any time.

Figure 4 shows sectional elevation of ammonia cylinder and head, while Figures 5 to 7 show details of ammonia cylinder, suction and discharge valves.

Figure 8 shows machine erected, and Figures 9 to 24 show parts of machine and machine in course of and method of erection.

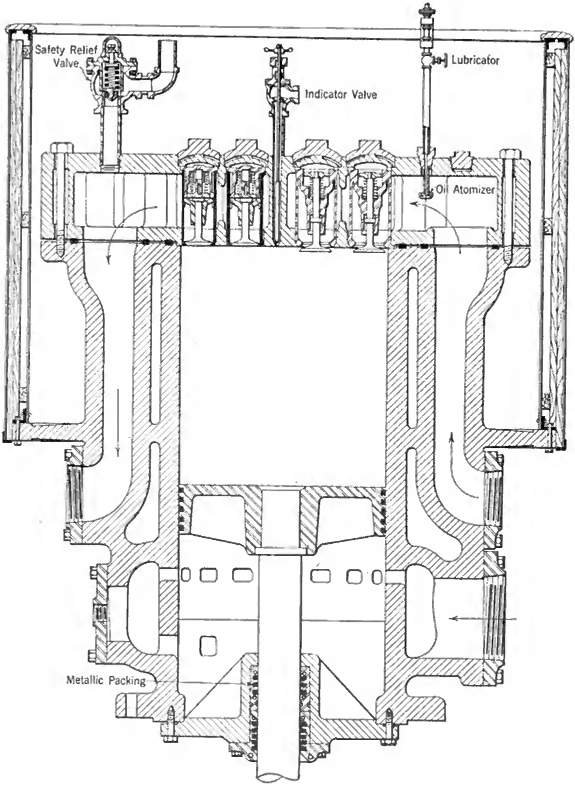

FIG. 4.—Sectional elevation of ammonia cylinder and head.

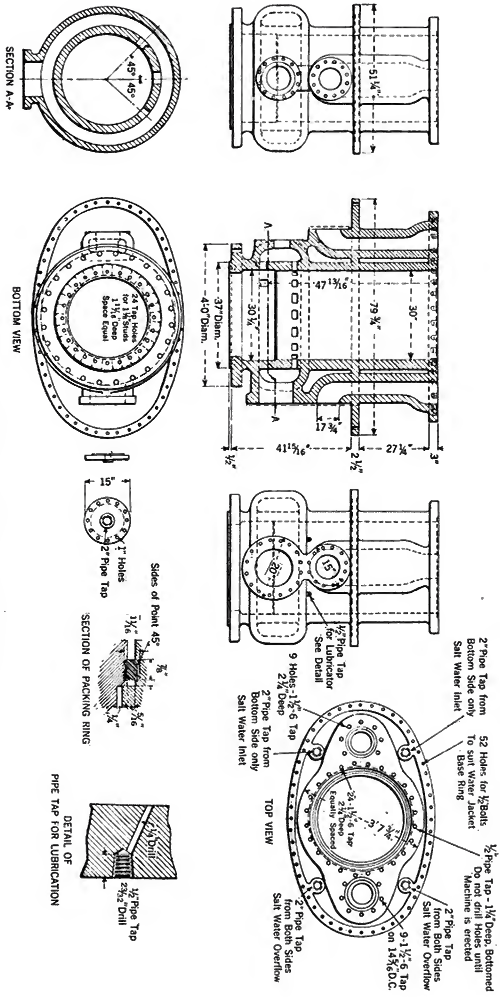
SECTION A-A
45°
45°
51¼"
BOTTOM VIEW
24 Tap Holes for 1⅛" Studs 1⅞" Deep Space Equal
A
A
47 13/16"
79¾"
30"
30½"
37" Diam.
4'-0" Diam.
17¾
41 15/16"
27¼"
½"
2½"
3"
15"
1" Holes
2" Pipe Tap
Sides of Point 45°
⅞"
1 1/16"
5/16"
¼"
SECTION OF PACKING RING
20"
15"
½" Pipe Tap for Lubricator See Detail
2" Pipe Tap from Bottom Side only Salt Water Inlet
52 Holes for ½" Bolts To suit Water Jacket Base Ring
9 Holes-1½"-6 Tap 2¼" Deep
2" Pipe Tap from Bottom Side only Salt Water Inlet
TOP VIEW
24-1½"-6 Tap 2¼" Deep Equally Spaced
3'-7¾"
½" Pipe Tap - 1¼" Deep. Bottomed Do not drill Holes until Machine is erected
2" Pipe Tap from Both Sides Salt Water Overflow
9-1½"-6 Tap on 14 5/16" D.C.
2" Pipe Tap from Both Sides Salt Water Overflow
¼" Drill
½" Pipe Tap 23/32" Drill
DETAIL OF PIPE TAP FOR LUBRICATION

FIG. 5.—Compressor cylinder, showing details.

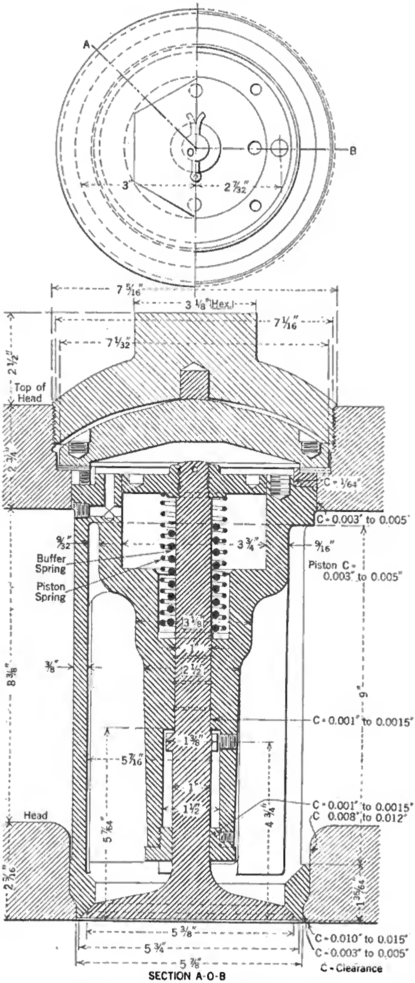

FIG. 6.—Suction valve and details.

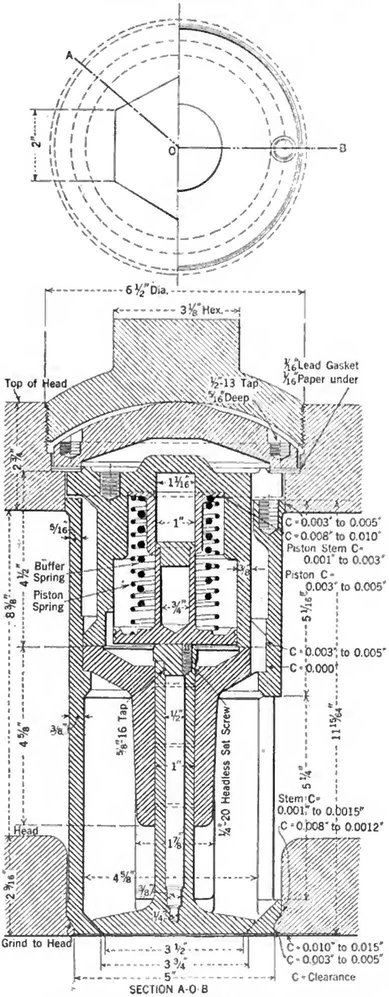

Fig. 7.—Discharge valve and details, showing cage and position in cylinder head.

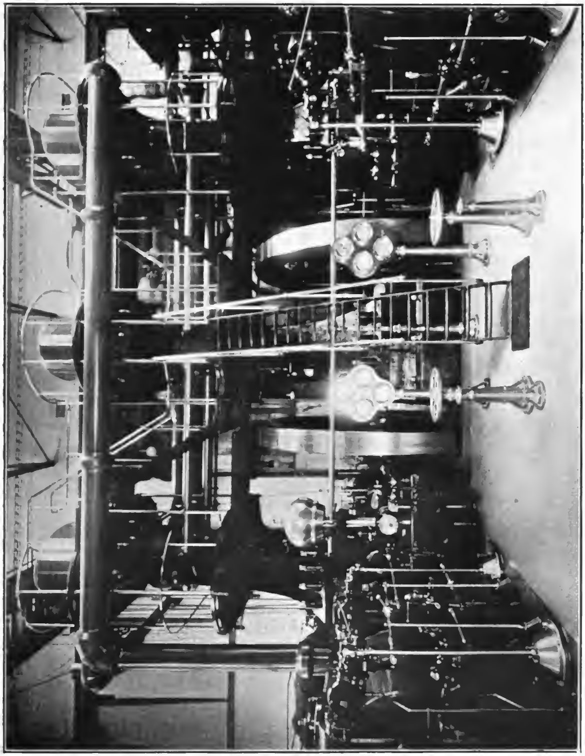

Fig. 8.—Completed machine running.

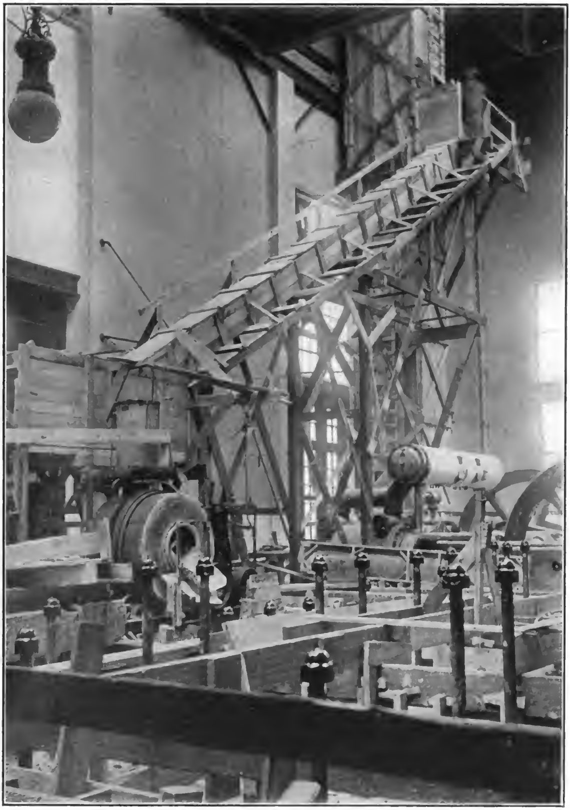

Fig. 9.—Side view of automatic concrete elevator, shute and mixer, for delivering concrete into foundation forms.

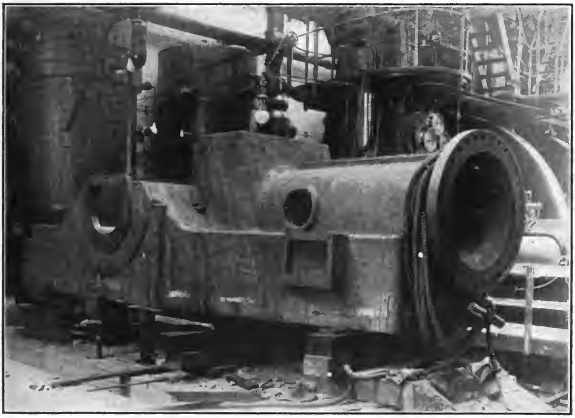

Fig. 10.—Low pressure bedplate lowered onto foundation, showing size as compared with the 800 ton machine in service in the background.

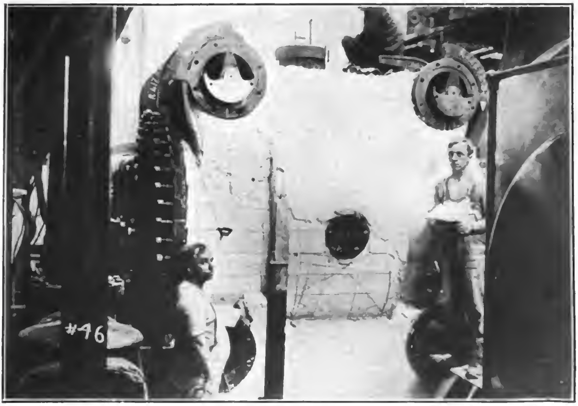

Fig. 11.—Low pressure cylinder, showing how insulation was carried forward with erection.

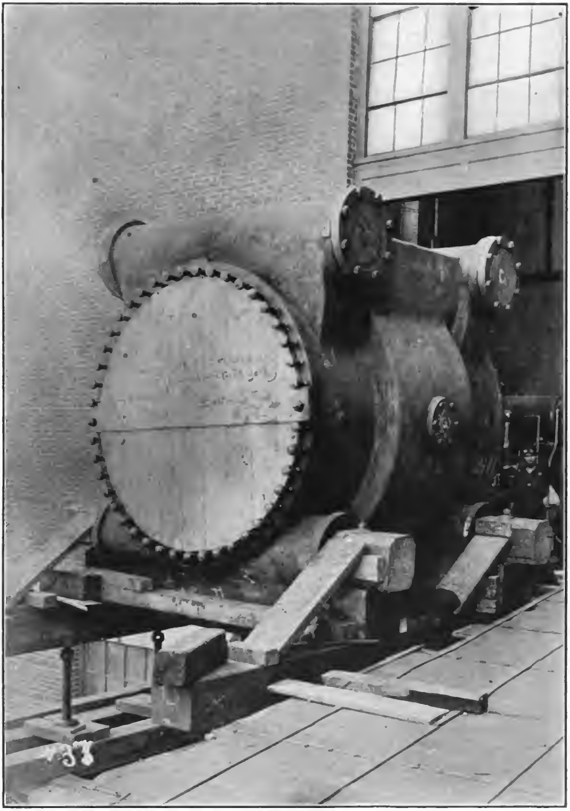

FIG. 12.—Low pressure cylinder being taken into power house.

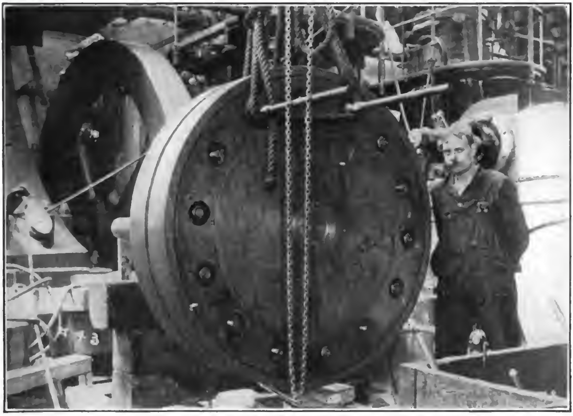

FIG. 13.—Low pressure piston.

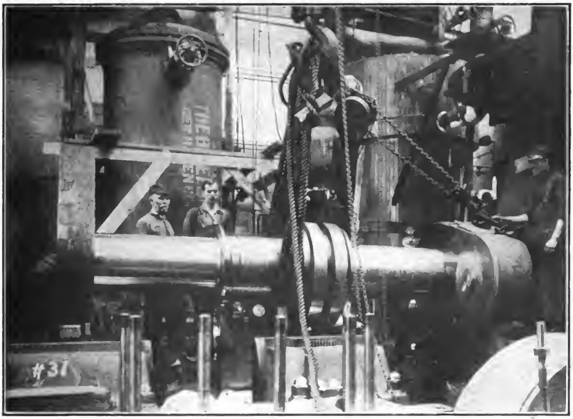

FIG. 14.—One section of the fluid compressed 24" main shaft being lowered into place. Forty-five per cent carbon, open-hearth steel forging, tensile strength 90,000 lbs. per square inch, elastic limit 50,000 lbs.

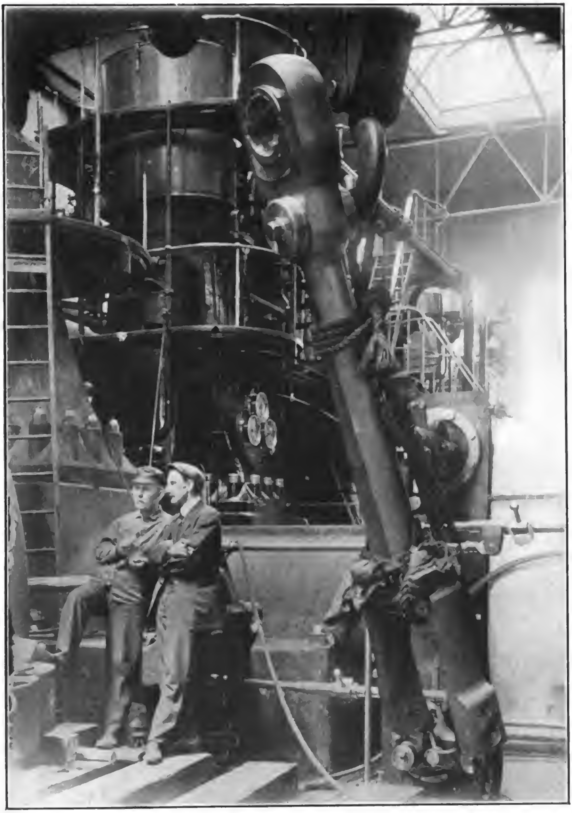

Fig. 15.—One of the main connecting rods.

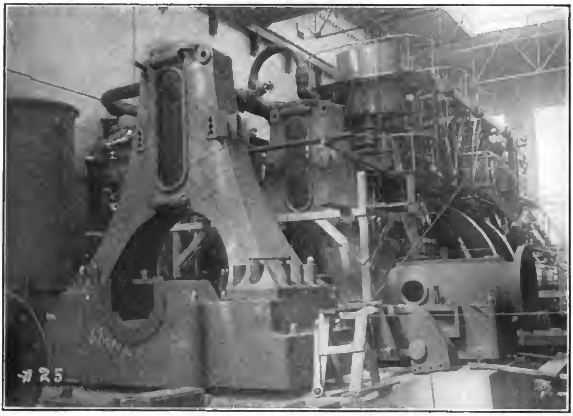

Fig. 16.—Low pressure and intermediate bedplates with A frames erected in place.

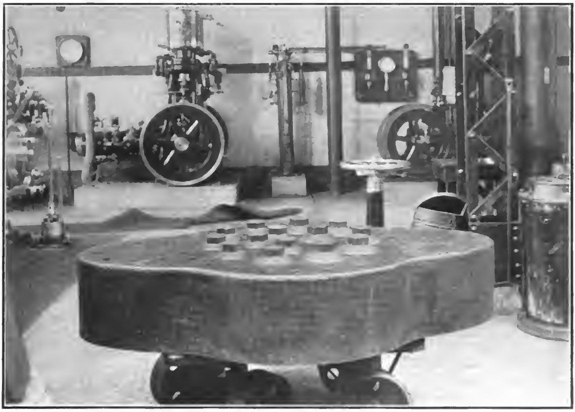

Fig. 17.—Ammonia cylinder head, showing removable valve caps.

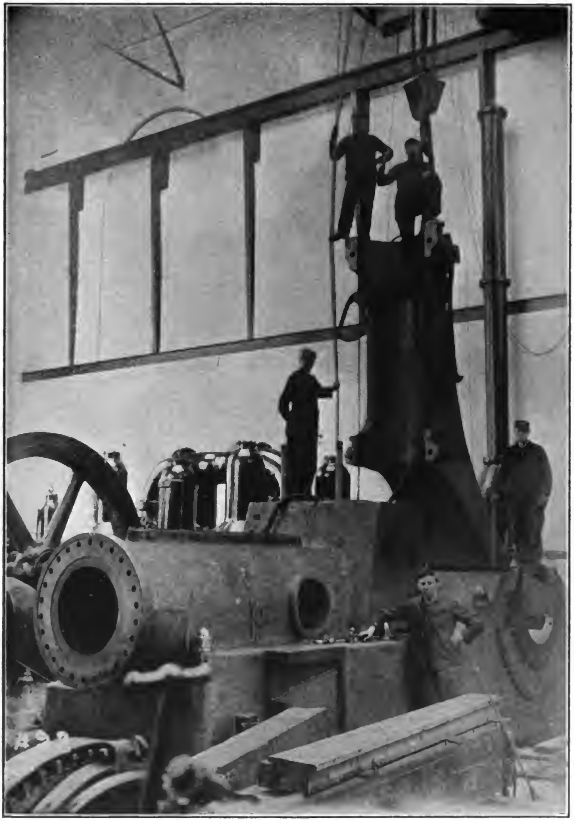

FIG. 18.—Low pressure bedplate, with one-half of the first A frame being erected.

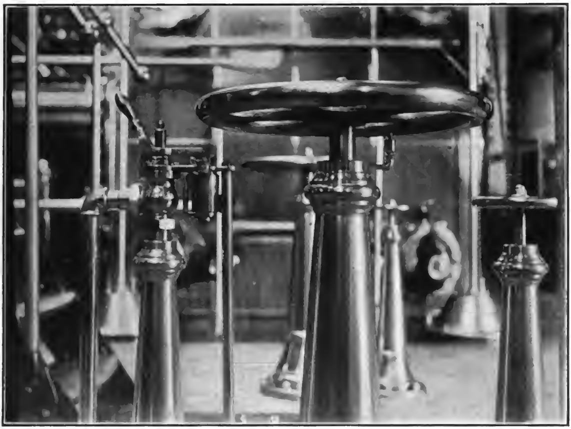

FIG. 19.—Steam valve stanchions, with mechanical and electrical quick-stopping device on the left.

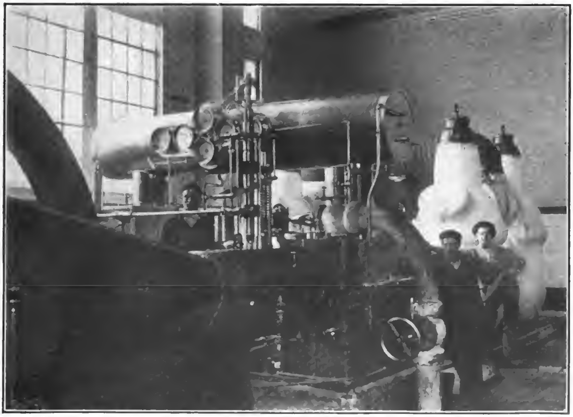

FIG. 20.—Ten million gallon brine pumping engine in service.

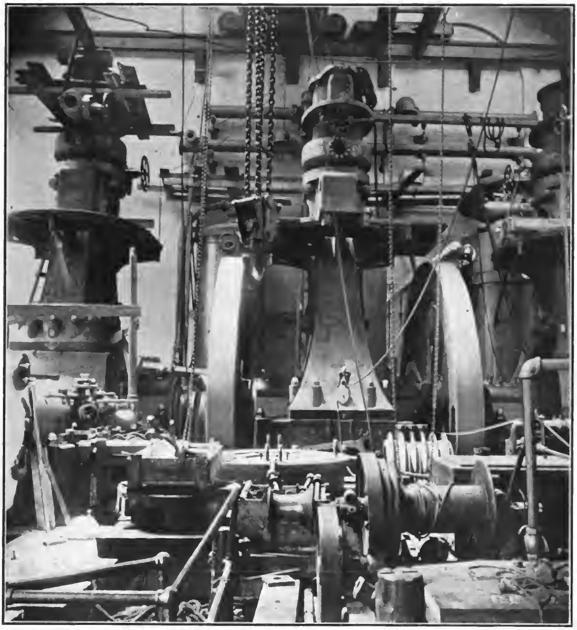

FIG. 21.—Skeleton of 1,000 ton machine, showing electric hoist in the foreground. Cranes and ammonia headers in the upper background.

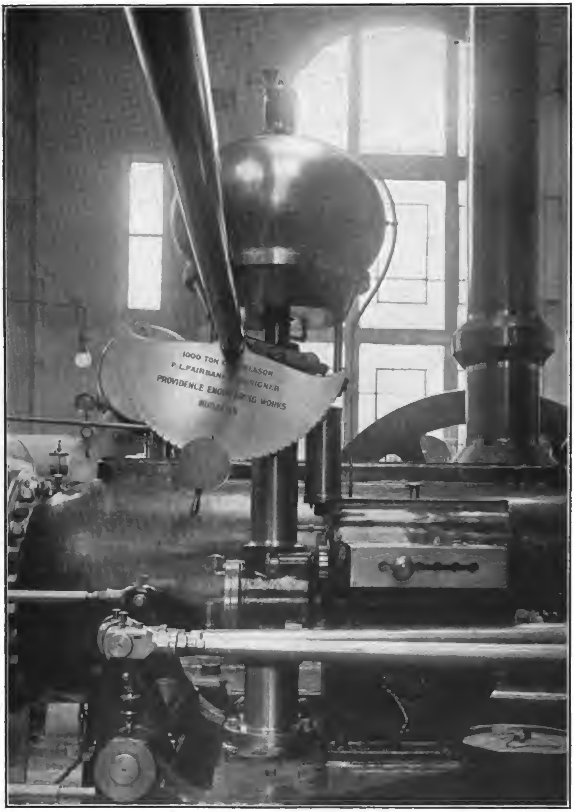

FIG. 22.—Governor, showing speed regulating pendulum with shifting gear transmission to control the speed of compressor from 15 to 75 r. p. m. by increments of one-third of a revolution, maintaining a practically constant speed of governor spindle. All parts being self-oiling and enclosed.

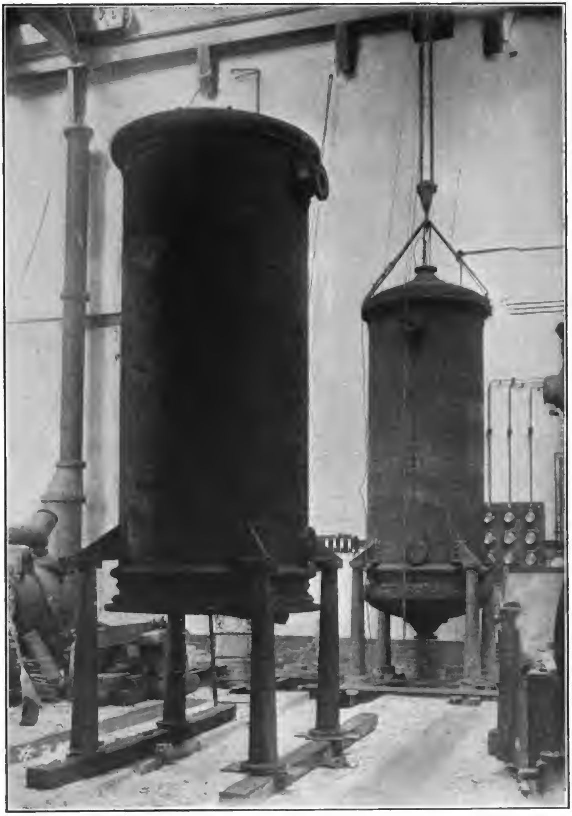

FIG. 23.—Two of the 200 ton brine coolers.

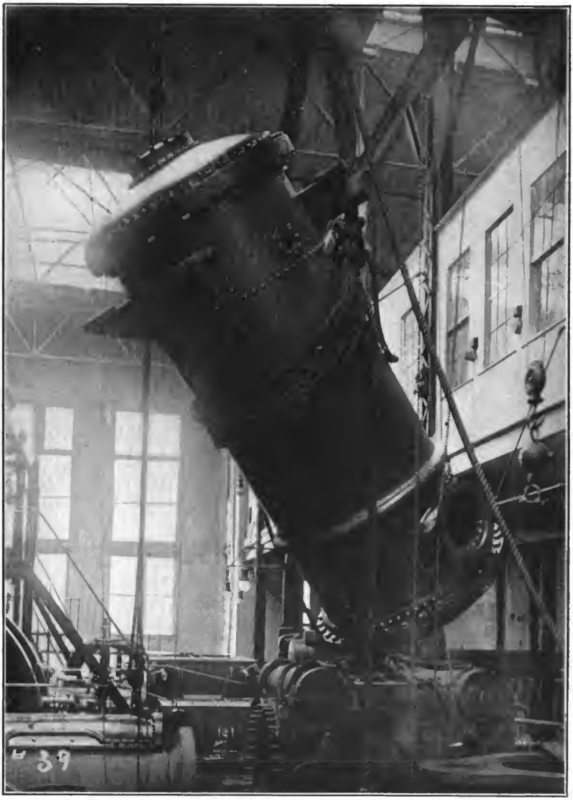

FIG. 24.—Exhaust steam receiver and reheater being lowered into place.

REVIEW

DECOMPOSITION OF AMMONIA

DECOMPOSITION OF AMMONIA. By Professor R. Plank, Danzig, Germany. *Zeitschrift für die gesamte Kälte-Industrie,* Berlin, March and May, 1915. 8 pp.

Prompted by recent discussions on the subject of decomposition of ammonia, the author has undertaken an analytical investigation based on the laws of reactions as known in chemistry.

Professor Plank agrees with the opinion generally held that all these cases of ammonia accidents were caused by instantaneous explosions and not by a slow combustion of the ammonia; ammonia being combustible only in pure oxygen and not in the ordinary atmosphere. To prove this contention the writer proceeds to investigate our present knowledge of the subject of disintegration of ammonia, quoting in his article a number of references to literature on the subject.

THE DECOMPOSITION OF AMMONIA AS AFFECTED BY CHEMICAL EQUILIBRIUM

The decomposition of ammonia proceeds according to the equation

$$2\,NH_3 = N_2 + 3\,H_2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

which means that by the complete decomposition of 2 parts of NH_3 by volume, 1 part by volume N_2 and 3 parts by volume H_2 result, thus doubling the total volume. With this reaction, as with every exothermic-chemical reaction, is connected the generation of a certain amount of heat, its amount depending upon the temperature. In the production of ammonia from the elements, known as the synthetic method[1], which is becoming more widely adopted, a quantity of heat is liberated, and for the decomposition of the ammonia a like quantity of heat must be supplied. According to Thomsen and Berthelot, at ordinary room temperature the heat required for the decomposition of 2 molecules (34 kg. NH_3) is 24,000 calories. This quantity changes with the temperature and the varying specific heats of NH_3, H_2 and N_2, it increases rapidly with increasing temperature, and at 850 degrees C. reaches 30,000 calories.

It is known that high temperature favors the decomposition of ammonia. At any temperature and pressure the thermo-dynamic equilibrium demands the presence of a definite quantity of each of the 3 constituents, although they may be represented in exceedingly small quantities. This theory of chemical equilibrium, which is also known as the statics of gas reaction, is now perfectly understood and accepted as correct.

Denoting the partial pressures of the several constituents in kg. per cm.^{2} by

$$p_{NH_3},\ p_{H_2} \text{ and } p_{N_2},$$

and the sum of the partial pressures by p, the absolute temperature in degree C by T, and the heat of the reaction with Q, we derive from the second law of thermo-dynamics the relation:

$$2 \ln \frac{p^3_{H_2}\, p_{N_2}}{p^2_{NH_3}} = -\int \frac{Q}{T^2}\, dT + C \ldots\ldots\ldots\ldots\ldots (2)$$

wherein C is a constant, the value of which is perfectly defined by Nernst's heat theorem.

According to Nernst,[2] the heat of disintegration of ammonia is the same in quantity as the heat of formation (which latter is negative and preceded by the minus sign), referred to 2 molecules NH_3, is expressed by the equation

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[1] To date by far the greatest quantity of ammonia was produced by the dry distillation of nitrogenous coal as a by-product in the manufacture of illuminating gas.

[2] Nernst, Zeitschr. f. Elektrochemie, 16, 1910, p. 100.

$$Q = -\,21{,}980 - 7.0\,T - 3.31 \times 10^{-3}\,T^2 - 2.44 \times 10^{-9}\,T^4 \ldots\ldots (3)$$

Inserting this value in equation 2, and employing common logarithms in place of hyperbolic logs, equation (4) is obtained:

$$\log.\ \frac{p^3_{H_2}\,p_{N_2}}{p^2_{NH_3}} = -\,\frac{21{,}980}{4.571\,T} + 3.5\ \log.\ T + {}$$
$$+\,\frac{3.31 \times 10^{-3}\,T}{4.571} - \frac{0.81 \times 10^{-9}\,T^3}{4.571} + 0.468 \ldots\ldots\ldots\ldots (4)$$

According to Nernst, the constant C is taken at 0.468 when the pressures are based on technical atmospheres (kg. per $cm.^2$); if the pressures are figured in physical atmospheres (760 mm. mercury) the constant becomes 0.44.

In place of the partial pressures in equation 4, it is found more convenient in practical calculations to base them on parts by volume (referred to the total pressure), meaning the fractional volume occupied by each gas in 1 cubic meter. For simplicity, denoting these volume-parts by the chemical signs of the respective gases, and remembering that these volume-parts are in the same relation as the partial pressures, namely:

$$p_{NH_3} : p_{H_2} : p_{N_2} : p = NH_3 : H_3 : N_2 : 1,$$

equation 4 becomes

$$\log.\ \frac{p^3_{H_2} \times p_{N_2}}{p^2_{NH_3}} = \log.\ \frac{H_2^3\,N_2}{NH_3^2} + 2\ \log.\ p,$$

and, therefore,

$$\log.\ \frac{H_2^3\,N_2}{NH_3^2} = -\,\frac{4810}{T} + 3.5\ \log.\ T + 0.724 \times 10^{-3}\,T -$$
$$-\,0.1774 \times 10^{-9}\,T^3 - 2\ \log.\ p + 0.468 \ldots\ldots\ldots\ldots\ldots (5)$$

From this equation may be derived the composition of the gas mixture as a function of pressure and temperature. The several volume-parts, NH_3, H_2 and N_2 herein, are not independent from one another. For example, assuming 1 cubic meter pure NH_3, of which x volume-parts NH_3 to have been decomposed under specified temperature and pressure, then the mixture still contains $1-x$ volume-parts of NH_3, while the result of the decomposition, according to equation 1, consists of $\frac{3\,x}{2}$ volume-parts, H_2 and $\frac{x}{2}$ volume-parts N_2. Thus from 1 cubic meter of originally pure NH_3 there results a mixture equal to

$$(1 - x) + \frac{3\,x}{2} + \frac{x}{2} = 1 + x \text{ cubic meter.}$$

Referred to 1 cubic meter the several volume-parts of the decomposed constituents are:

$$NH_3 = \frac{1 - x}{1 + x},\ H_2 = \frac{3\,x}{2(1 + x)},\ N_2 = \frac{x}{2(1 + x)}.$$

Denoting the ammonia content with y, by placing $\frac{1 - x}{1 + x} = y$, the following equation is obtained:

$$\log.\ \frac{H_2^3\,N_2}{NH_3^2} = \log.\left(\frac{27}{16}\ \frac{x^4}{(1 - x^2)^2}\right) = \log.\left(\frac{27}{256}\ \frac{(1 - y)^4}{y^2}\right) =$$
$$= -\,0.9768 - 2\ \log.\ \frac{y}{(1 - y)^2} \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Inserting this value in equation 5 we obtain:

$$\log. \frac{y}{(1-y)^2} = \frac{2405}{T} - 1.75 \log. T - 0.362 \times 10^{-3}\, T + $$
$$+ 0.0887 \times 10^{-6}\, T^2 + \log. p - 0.722 \ldots\ldots\ldots\ldots\ldots (7)$$

From this equation the ammonia content y may be computed in volume-parts per cubic meter, or 100 y in per cent, for the condition of equilibrium. Since at very high temperatures practically all ammonia is decomposed, resulting in a very small value for y, a close approximation is obtained by using in place of equation 7 equation 8:

$$\log. \frac{y}{(1-y)^2} = \backsim \log. y \ldots\ldots\ldots\ldots\ldots\ldots (8)$$

In equation 7 the members T and T^2 are merely correction factors, it is evident, therefore, that the ammonia content y corresponding to the chemical equilibrium will be the greater the lower the temperature and the higher the pressure. Accordingly, the decomposition of ammonia is always hindered by a lowering of the temperature and by an elevation of the pressure, which latter fact will be recognized to be contrary to the erroneous opinion generally held. In the state of equilibrium less ammonia will be decomposed at high pressure, although the method of pressure increase may, for kinetic reasons, favor the decomposition, as will be explained further on.

The ammonia content 100 y in per cent has been calculated according to equation 7 for various pressures and temperatures, and the values are given in Table I and plotted in Figs. 1 and 2.

Table I.

t in deg. C = T in deg. C =		—20° 253	0° 273	+50° 323	100° 373	150° 423	200° 473	300° 573	500° 773	700° 973
p in atmospheres, absolute		%	%	%	%	%	%	%	%	%
	1	99.43	98.65	92.53	75.5	47.0	20.8	2.73	0.125	0.0178
“	4	99.72	90.32	96.20	86.80	68.1	43.0	9.46	0.50	0.0712
“	7	99.78	99.49	97.11	89.93	74.75	52.5	14.70	0.87	0.125
“	10	99.82	99.57	97.58	91.45	78.35	58.1	18.95	1.225	0.178

From Table I, as well as from equations 7 and 8, it will be seen that at high temperatures the small ammonia content y is directly proportional to the pressure. For very large values of y, however, which means at temperatures below 0° C., the decomposed quantity $z = l - y$ (referred to 1 cubic meter after decomposition), becomes very small. We then have in equation 7:

$$\log. \frac{y}{(1-y)^2} = \log. \frac{1-z}{z^2} = \quad - 2 \log. z \ldots\ldots\ldots (9)$$

with the result that then the decomposed quantity of ammonia is inversely proportional to the square root of the pressure. This agrees exactly in the case of the tabular values at $t = - 20°$ C. The values of y calculated by equation 7 agree very closely with the direct measurements made by Jost in Nernst's laboratory. But for reasons mentioned further on the comparison between theory and test is possible only at high temperatures.

In the case of 1 kg. per cm² pressure, the following comparison obtains wherein, in view of the very small ammonia content, equation 8 was used:

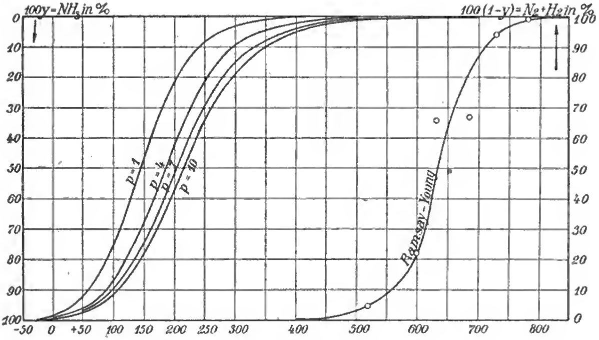
100y=NH₃ in %
100(1-y)=N₂+H₂ in %
0
10
20
30
40
50
60
70
80
90
100
100
90
80
70
60
50
40
30
20
10
0
-50
0
+50
100
150
200
250
300
400
500
600
700
800
p=1
p=4
p=7
p=10
Ramsay-Young

FIG. 1

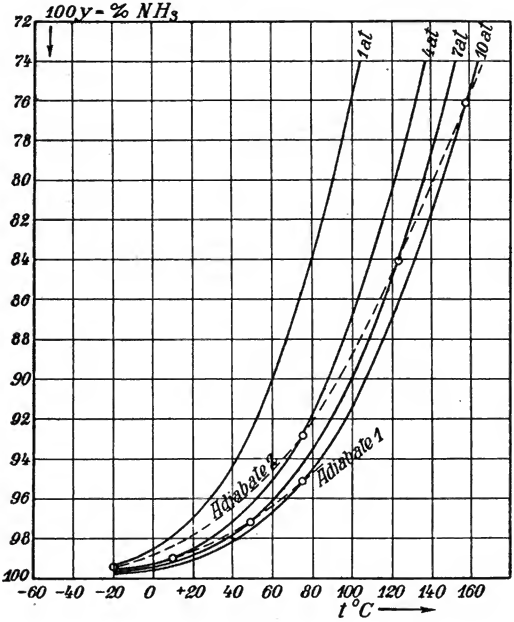
100y - % NH₃
72
74
76
78
80
82
84
86
88
90
92
94
96
98
100
1at
4at
7at
10at
Adiabate 2
Adiabate 1
-60
-40
-20
0
+20
40
60
80
100
120
140
160
t °C

FIG. 2

TABLE II

Absolute Temp. T	y calculated by equ. 7 & 8	y as determined by Jost.
973°C.	0.0178%	0.0169%
1073	0.0086	0.0085
1203	0.0041	0.0042
1273	0.0028	0.0031

It is seen that the agreement is very close. It may be added that within the temperature limits mentioned exactly the same quantity of NH_3 is obtained, whether beginning with pure ammonia, which becomes greatly decomposed, or whether the initial gas mixture consists of $N_2 + 3 H_2$. Furthermore, the tests have definitely proven that the ammonia content with very small values of y increases in proportion to the pressure. This fact is of particular importance in the technical production of ammonia by synthetic means, in which, as a matter of fact, pressures of several hundred atmospheres are used.[3]

Equation 7 shows that at every final temperature and every final pressure all three constituents of the chemical reaction are represented in the mixture. Fig. 1, however, shows that at temperatures below 0° C., especially at high pressures, the quantity of ammonia decomposed is exceedingly small, while at temperatures above 700° C. there remain merely traces of ammonia. Long before this exact theory had been established, Ramsay and Young[4] had pointed out that the decomposition of the ammonia even at the highest temperatures was never complete. At the more recent tests by Perman and Atkinson, Jost, Haber and Le Rossignol, measurable quantities of ammonia were found even at temperatures as high as $t = 1.100°$ C.

Fig. 1 shows that the decomposition of ammonia increases greatly between $t = 100°$ and 300° C. Fig. 1 also enables us to make certain deductions regarding the behavior of ammonia in respect to fire hazard and loss of capacity in refrigerating machines. These deductions, which are based on the statical theory of chemical equilibrium, will further be found to be greatly modified and improved by the consideration of the effect of kinetic conditions which, unfortunately, are as yet not entirely understood. One of the questions involved is the velocity of reaction and the catalytic action of solid bodies, such as metallic walls, with which the reaging gases come in contact. But here, also, pressure and temperature qualitatively exert the same influence upon the decomposition, so that the equations arrived at so far still apply, but on a different scale.

In mechanical refrigeration we are concerned with the temperature range, including about —50° and —150° C. at pressures of 1 to 14 atmospheres. absolute. The low pressures, 1 to 4 atmospheres, absolute, and temperatures —50° to 0°, apply to the suction side of ammonia compressors, while the higher pressures, 7 to 14 atmospheres, absolute, and temperatures 30° to 150° C., apply to the discharge or high-pressure side of the system. According to our previous reasoning it is fortunate, so far as decomposition is concerned, that the high temperatures are associated with high pressures. The temperature range which refrigerating engineers are particularly interested in has been cut out of Fig. 1 and reproduced on a larger scale in Fig. 2. This diagram shows that on the suction side the decomposition of ammonia, even under unfavorable conditions, will not exceed 1 per cent, whereas, on the discharge side, the decomposition may proceed much farther. We will calculate two examples to see how the decomposition increases during compression in the cylinder, assuming that the chemical equilibrium is maintained at every point of the compression curve, which requires, of course, high velocity of reaction.

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[3] Haber and Le Rossignol, Zeitschr. f. Elektrochemie, 19, 1913, p. 53.
[4] Ramsay-Young, Journal Chem. Soc., 45, 1884, p. 88.

Example 1. Evaporator pressure, 4 atmospheres, absolute (422 pounds per square inch gauge). Evaporator temperature, —2° C. (28° F.). Condenser pressure, 10 atmospheres, absolute, (128 pounds per square inch gauge). Condenser temperature, + 24° C. (75.2° F.).

p Atmos. Absol.	T	t		y
4	283°C.[5]	10°C.	50°F.	99.0%
7	322	49	120	97.2
10	349	76	169	95.2

The values for y are plotted in Fig. 2 and marked "Adiabate 1."

Example 2. Evaporator pressure, 1 atmosphere, absolute, (— 0.5 pound per square inch gauge). Evaporator temperature, — 32° C. (— 26° F.). Condenser pressure as before (128 pounds per square inch gauge).

p Atmos. Absol.	T	t		y
1	253°C.[5]	—20°C.	—4°F.	99.43%
4	348.5	+75.5	168	92.8
7	396	123	253	84.1
10	430	157	315	76.2

These values for y are also plotted in Fig. 2 and connected by a curve marked "Adiabate 2."

In Fig. 2 the favorable hindering influence of the pressure increase upon the decomposition is plainly discernible. Even then, there appears, with dry compression under most favorable conditions, a decomposition amounting to about 5 per cent, which under extreme conditions increases to 25 per cent, compared with the condition of equilibrium.

But on the way to the condenser, and in the condenser, the gases become cooled to the liquefaction temperature, whereby a great amount of the ammonia is recovered. Thus, at a condenser pressure of 9 atmospheres, the saturated vapor of 21° C. contains again 99 per cent NH_3. The height of the condenser pressure is of scarcely any influence in this connection, because it is offset by the counter action of the condensing temperature. For example, in Table III we obtain:

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[5] The temperature at start of compression has been taken higher than the evaporator temperature, owing to the absorption of heat by the vapor in the suction pipe and cylinder walls.

TABLE III

Condenser pressure p		Condenser Temperature			Ammonia content y per equ. 7 & 9.
Atmos. Abso.	Lbs., sq. in., gauge	T			
7	84.8	286° C.	13° C.	55.7° F.	99.14%
8	99.1	290	17	62.9	99.07
9	113.3	294	21	69.5	98.99
10	128.	297	24	75.4	98.93
11	141.7	300	27	80.9	98.87
12	156.1	303	30	86.3	98.81

It is seen that under very high condenser pressures the quantity remaining decomposed is a trifle greater. Since this decomposed part consists of gases (H_2, N_2) difficult to liquefy, this part will not condense; it will be carried as ballast through the condenser and evaporator. But as the quantity does not exceed more than about 1 per cent, the loss is not serious. The decomposed gases may become dissolved largely in the liquid ammonia.

Fig. 2 also indicates that with wet compression the amount of ammonia decomposed during the compression is the less the lower the superheat at end of compression. In the condenser and evaporator, however, there is no change.

THE DECOMPOSITION OF AMMONIA AS AFFECTED BY KINETIC CONDITIONS

So far we have concerned ourselves only with the chemical equilibrium of gas reaction, equation 1, and have found that at temperatures of about $t = 100°$ C., as found in refrigerating machines, a perceptible quantity of ammonia should decompose into its elements. In the same way there ought to be formed a considerable amount of ammonia at these temperatures from a mixture of $N_2 + 3\,H_2$. In contradistinction to this theory, the actual test proves that there is neither a decomposition of the pure ammonia, nor a formation of the same out of the elements at low temperatures, and that the condition of equilibrium is not reached in either case. The cause for this lies in the fact that the equilibrium is obtained exceedingly slowly at low temperatures, owing to the effect of the exceedingly inactive nitrogen.[6] Hence the velocity of reaction is here almost zero; increases, however, rapidly as the temperature is increased. A mixture of $N_2 + 3\,H_2$ at room temperature is chemically indifferent, not because of the absence of chemical affinity, but because the velocity of reaction is exceedingly small. Physico-chemists do not doubt that, for example, a mixture of hydrogen and oxygen will at ordinary temperatures change gradually to water, only the quantities of water formed during the time available to the experimentors are not great enough to be measurable. Evidently the determination of the decomposition of gas mixture does not only depend upon the equilibrium, but also upon the velocity with which this equilibrium is reached.

It has been found that in the case of different gas reactions there are certain solid bodies, in the presence of which the velocity of reaction increases, without these bodies suffering any chemical change. This phenomenon is known as catalytic. For example, a catalytic action is exerted by the walls of a vessel containing reaging gases. In the free gas space the velocity of reaction is always smaller than at the walls. Our knowledge of these catalytic gases is as yet incomplete.

We will briefly review previous investigations on the decomposition of ammonia as affected by kinetic conditions. To my knowledge Ramsay and

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[6] In the production of oxide of nitrogen, NO, from air it is found that the nitrogen reages very slowly even at 2000 deg. C.

Young (1884) were the first to investigate this subject. They operated in the temperature interval of $t = 500°$ C. and about $830°$ C., and proved that the decomposition of ammonia under the most favorable circumstances commenced but slightly below $500°$ C. But to-day we are positive that for every temperature below $500°$ C. there must be decomposition of ammonia which, however, escaped the observers within the short testing periods at their disposition, because the velocity of reaction is so slow. Ramsay and Young studied the decomposition inside of tubes made of various materials, and in addition introduced in the reaction vessel various solid substances. Tests were made with a tube of porcelain and a tube of iron, both tubes containing broken pieces of porcelain. It was found that the decomposition in the iron tube was one and one-half to two times greater than in the porcelain tube. According to Fig. 1 the equilibrium at $t > 500°$ favors $N_2 + 3\ H_2$. On Fig. 1 have been plotted the quantities of ammonia found by Ramsay and Young in the iron tube, assuming that they started with pure ammonia. It is seen that the decomposition reaches completion at $t = 780°$ C. only. Hence beyond this temperature only the time of reaction in comparison to the time of observation may be neglected. These observations agree very well with the recent determination by Haber,[7] who found that the velocity of reaction at $t = 700°$ C. was still so small that the equilibrium could not be reached with certainty.

Comparing in Fig. 1 the curve of Ramsay and Young's values with the curve representing equilibrium at 1 atmosphere pressure ($p = 1$), it is seen that the two curves are very similar, only one curve is moved to the right about 450 to 500° C. Accepting this transition of the curves, the quantities of ammonia actually decomposed at low temperatures could be approximated by means of equations 7 and 9, the resulting values becoming very small, extending the duration of decomposition over months and years. The conclusion would be reached that the decomposed quantities are too small to be of any influence in the working of the refrigerating machine if it were not for another circumstance, the exact influence of which is beyond calculation, but tends to accelerate the decomposition. We will revert to this further on.

The tests by Ramsay and Young proved that the decomposition of NH_3 progresses slower in the porcelain than in the iron tube (iron being a better catalysor), but their tests with a glass tube showed a still slower decomposition, becoming noticeable only at 780° C. Evidently the glass surface effectively hinders the decomposition.

Perman and Atkinson[8] also made tests, working between temperatures of $t = 677$ and $1{,}111°$ C., again using a reaction vessel made of porcelain. The quantity of decomposed ammonia was measured by the resulting pressure increase at constant volume (it being remembered that the decomposition doubles the volume at constant pressure). The investigators proved that the decomposition was accelerated by the presence of certain metals, such as mercury, iron and platinum. Their tests are difficult to interpret, because no two tests were made under comparative conditions. In each case a number of variables were introduced by changing at once the temperature, the pressure, at start, and the catalysor. The one positive result which may be based upon these tests is the fact that the reaction in the main takes place at the surface of the solid bodies and not in the gas space. Each sort of solid catalysor causing a different reaction, the reaction within the mass of gas would necessarily be insignificant.

The third important work was done by Bodenstein and Kranendieck,[9] who operated between 780° and 880° C. They used quartz glass for the catalysor and the reaction vessel, and measured the decomposed gases by a manometer after the method of Perman and Atkinson. Unfortunately the material they used renders interpretation of their splendid and systematic tests very difficult. The observers reached the conclusion that the reaction takes place not so much on the smooth surface of the quartz glass, but on the far greater surface of the pores in which the ammonia is adsorbed. By this they explain the relatively long times of reaction and the very slow increase of reaction velocity at increasing temperature. At high temperatures the velocity of the chemical reaction, equation 1, is doubtless very great. But the whole process of decomposition is greatly impeded by the low adsorption velocity in the pores, also the diffusion of the undecomposed ammonia from the gas space to the quartz surface, as well as the counter diffusion of the products of decomposition through the adsorbing layer.

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[7] Haber and Le Rossignol, Zeitschr. f. Elektrochemie, 14, 1908, p. 194.

[8] Perman and Atkinson, Proc. of the Royal Soc., 74, 1905, p .110.

[9] Bodenstein and Kranendieck, Festschrift, W. Nernst gewidmet, 1912 (pub. by Knapp, Halle), also Kranendieck, dissertation, Hanover, 1912.

The tests by Perman and Atkinson have proven the accelerated action of certain metals upon the decomposition of ammonia, while the tests of Haber and Le Rossignol[10] showed that the formation of ammonia also was accelerated by the presence of metals, such as manganese, iron and especially osmium and uran. Clearly iron accelerates the approach to equilibrium in both directions. But all catalysors mentioned are effective only at high temperatures. In this connection a communication by Lipski[11] is of importance, according to which Cer (Cerhydride or Cernitride) is a catalysor, showing its greatest activity in the synthetic manufacture of ammonia in the temperature range $t = 200$ to 300° C. Haber and Le Rossignol,[10] however, were unable to confirm this property of Cer.

CONCLUSION

In view of all that has preceded, it may be said that the slowness with which the reaction, equation 1, proceeds at low temperatures appreciably modifies the purely statical observations made in the first half of this article. Qualitatively the influence of temperature and pressure variations remain, but quantitatively the low reaction velocities cause a considerable shifting of the resulting gas composition in the direction of low temperatures. This fact is of great advantage in mechanical refrigeration, because a decomposition of the ammonia during the compression would be accompanied by a corresponding increase in volume, and therefore an increase in power consumption; on the other hand, we have seen that the refrigerating capacity itself is scarcely affected by the decomposition, even at the point of equilibrium, because of the cooling effect in the condenser, which allows at most 1 per cent to remain decomposed.

But for the synthetic manufacture of ammonia the low reaction velocity is a disadvantage. We are compelled to place the reaction at a higher temperature level where nitrogen and hydrogen remain stable, leaving only traces of ammonia to be obtained.

The result now arrived at does, however, not quite agree with actual practice, because cases are known where large quantities of hydrogen and nitrogen have been found in the mains of ammonia refrigerating machines.[12] The question is, how did this decomposition take place? No doubt in every case the compressors were operated with a high degree of superheat. Hence, we must inquire what other influences outside of the catalysor might favor decomposition. These influences include:

1. Effect of heat (temperature increase); this has already been fully discussed.
2. Photo-chemical effects (radiation by short-waved chemically active rays).
3. Electro-chemical effects (cathode rays, quiet electrical discharge, Siemen's tubes, arc-light discharge).[13]
4. Mechanical impact (sudden compression).

Of the effects enumerated, only the first and the fourth apply in mechanical refrigeration. The exceedingly short time (0.2 to 0.05 seconds) available for the compression of the ammonia vapors, especially in high-speed machines, may greatly favor the decomposition of ammonia. The extent of the effect, however, cannot be calculated. At any rate, the rapid compression may be considered as equal to mechanical impact, and therefore the decomposition in the compressor cannot be compared with that obtained in laboratory vessels.

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[10] Haber and Le Rossignol, Zeitschr. f. Elektrochemie, 19, 1913, p. 54.

[11] Lipski, Zeitschr. f. Elektrochemie, 15, 1909, p. 189.

[12] A. S. R. E. Journal, May, 1915, p. 40.

[13] Regarding the electro-chemical equilibrium of ammonia, see especially Davies, Zeitschr. f. Physik. Chemie, 64, 1908, p. 657.

Acceleration of the decomposition by the compression impact appears to be the only possible explanation for the noticeable quantities of products of decomposition found in practice. In this case the decomposition is objectionable, because the quantities once decomposed are not recovered because of the slow reaction velocity obtaining also in the formation of ammonia, even if their temperature by the action of the cooling water in the condenser remains unchanged. They reduce the refrigerating capacity and increase the power consumption until they are expelled from the system.

Wherever the condenser pressure is unduly high, as in cases of very low temperatures and unfavorable conditions of cooling water, and where with dry compression high temperatures of superheat are to be expected, it will be advisable to resort to means for ameliorating the decomposition. The following expedients are suggested:

1. Avoidance of high speed.

2. Abandonment of completely dry compression; adjusting of the feed valves in such manner that the temperature in the discharge pipe will be at most 80° to 100° C. (176° to 212° F.).

3. Effort to reduce to a minimum the time of exposure to high temperature. This may be accomplished by water-jacketing the discharge pipe right at the compressor cylinder to promptly remove the superheat. A cylinder of sheet metal around the discharge pipe with cooling water flowing in opposite direction to the gas should meet the requirements.

4. Improving the catalytic action of the surfaces enclosing the hot gas. We have seen that iron favors the decomposition of ammonia, while glass hinders it. Hence, it would seem justified to line the discharge pipe in its hottest section with enamel or some similar coating.[14] No such expedients can be applied in the compressor cylinder, but by proportioning diameter and stoke in such a way that a minimum of surface is obtained, an improvement will be gained also from the viewpoint of diminishing cylinder heating.

The author is of the opinion that in no case are the quantities of decomposed ammonia in the system sufficient to form, with the atmosphere, an explosive mixture. He holds that the explosions reported can only have occurred by great quantities of undecomposed ammonia escaping from the machine. In striking open lights these became sufficiently heated to cause high reaction velocities, thus causing a complete decomposition of the ammonia *outside* of the machine. The resulting great quantities of hydrogen becoming mixed with the oxygen of the air caused the explosions. It was shown in the first part of this article that complete decomposition of the ammonia, after escape into the atmosphere, was accomplished much more readily than inside the machine, because of the very low partial pressure.

The same condition holds when an open light is introduced into a compressor. The flame induces the decomposition, and therefore open flames should be prohibited in the neighborhood of ammonia machines.

The writer agrees with M. Krause, who is of the opinion that in the tests of Schlumberger and Piotrowski,[15] the electrical spark caused local decomposition of the ammonia. The combustion of the hydrogen raised the temperature of the remaining mass of gas to that corresponding to high reaction velocities, effecting complete decomposition of the entire quantity.

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[14] Enameled Pipe & Engineering Co., Elyria, O.

[15] Journal f. Gasbeleuchtung und Wasserversorgung, 1914, p. 941.

GAS PRODUCERS WITH BY-PRODUCT RECOVERY

Gas Producers with By-Product Recovery. By Arthur H. Lymn, London, England. Presented at the New York local section of the American Society of Mechanical Engineers, November 10, 1914. Journal of the A. S. M. E., May, 1915. 13 pp.; 22 Figs.

A historical resumé of the development of the producer-gas generating art is given, beginning with an illustrated description of Young & Beilby's by-product gas producer, by which the inventors claimed as early as 1883 to recover in the form of ammonia from 60 to 70 per cent of the total nitrogen in the fuel. In the same year Dr. Ludwig Mond patented his well-known by-product recovery process, which was put into commercial use about 1890.

In both the Young and the Mond producers distillation took place in a downward direction in the upper part, the coke residue being gasified in the lower part, the gases mixing and leaving the producer together.

In the Quintin Moore system the ammonia yield was claimed to be increased with only about half the amount of steam in the air blast, by a design in which the producer was divided into three parts—a lower brick-lined part, a middle water-jacketed part and an upper air-cooled part.

In the Lymn system effort was made to employ the advantages of the former types without their disadvantages. In this producer the cumbersome tile-packed tower or horizontal dasher washers were replaced by vertical washers, in which intensive washing was realized, principally through the momentum of the gases. This type of washer first had a series of co-axial revolving discs, the top one of which sprayed the liquid into a collecting cone, which delivered it into the next disc below, but when it was found that the movement of the gas was more effective than that of the discs, this mechanical feature was eliminated. In the Lymn system exactly the same kind of apparatus is employed for absorbing the ammonia and recovering the steam as for washing the gas. To remove final traces of tar, centrifugal cleaners and dry scrubbers, such as proposed by H. F. Smith, are employed. Another improvement claimed by the author is means of removing much of the dust in a dry state by means of cyclone dust separators of special design. Mechanical agitation of the ash and fuel zones and mechanical removal of the ash were employed.

Among the interesting data given in tabular form may be cited the following, derived from actual operating results of a Lymn system:

	First 4 Weeks	Second 4 Weeks
Yield of sulphate of ammonia per ton coal..	60 lbs.	61 lbs.
Yield of tar (containing water) per ton coal.	230 lbs.	257 lbs.
Sulphur contained in gas (average).........	0.63	0.38 grams per cu. m.
Tar contained in gas (average)............	0.04	0.057 " " "
Nitrogen efficiency during both periods......		70 per cent

In conclusion, the author calls attention to the importance of the sulphate of ammonia industry, showing by means of curves that between 1900 to 1913 the value of this commodity imported into the United States ranged from a minimum of $500,000 to $6,500,000 per year, at a cost of from $55.00 to $67.00 per ton. The more than 560,000,000 tons of coal produced per annum in the United States contains on an average of 1.3 per cent nitrogen in the form of ammonium sulphate, would have a value of $600,000,000.

ICE MAKING BY CENTRAL STATION POWER

ICE MAKING AS A BY-PRODUCT OF CENTRAL STATIONS. Haywood Cochran, Chicago, Ill. Journal of the American Society of Mechanical Engineers, July, 1915. 6 pp.

In his paper read before the Chicago local section of the A. S. M. E. the writer describes the compression and absorption types of refrigerating machines and shows how they can be operated in connection with central stations. These ice plants may be operated as the property of the central station, adjacent to which they are usually located, in which case exhaust steam absorption machines may be employed, or as privately-owned plants located at points most favorable for the distribution of ice, in which case compression systems are usually employed, driven by electric current or steam purchased from the central station.

Where absorption machines are employed and water is available at 70° F., the plant can be so designed as to require an exhaust steam pressure of only 3 pounds in the generator. About 55 to 60 pounds of steam per hour per ton of ice is required for this purpose. With from 90° to 95° condensing water it is not possible to run on less than from 20 to 25 pounds exhaust steam pressure because of the resulting high condenser pressure.

It is claimed that when properly designed the absorption machine is ideal for warm water conditions, because it takes comparatively little more steam in the generator to produce 200 pounds than it does to produce 150 pounds pressure, just as it takes little more coal to carry a boiler pressure of 125 pounds than it does for 100 pounds (less than 1 per cent).

An electrically-driven compression plant will require from 43 to 70-kilowatt hours per ton of ice per day, depending upon its size and other conditions. At 1 cent per kilowatt-hour the power cost per ton averages between 50 and 60 cents. Table 2 of his paper shows the operating cost of compression machines using various sources of power.

From the central station manager's standpoint the electrically-driven, privately-owned ice plant is preferable, but in smaller cities it is more profitable for the central station to control the ice plant.

From the ice plant manager's standpoint the advantages of purchasing power at a cost of 50 cents a ton or over is very questionable. Tabulated records of the Muncie Ice & Coal Co., Muncie, Ind., give an average ratio of 8.86 pounds of ice per pound of coal burned and an average fuel cost per ton of ice of $0.203, which, plus fireman's wages, interest and depreciation on the boiler plant, should not exceed $0.35. In a properly designed combination plant consisting of a 20-ton compression and a 60-ton absorption unit, for example, a still lower fuel cost per ton of ice is possible. In this case the compression machine would be used on one-half of the coils in one of two 40-ton tanks and the absorption machine on the remaining coils. The compressor engine and auxiliaries would furnish the 3,600 pounds of steam per hour required by the absorption machine generator. In a raw water plant, in which an evaporation of 6 is obtained with $2.00 coal, the fuel cost per ton of ice would be $0.16. Such an 80-ton plant should be operated with one 125-horsepower boiler with another of the same size in reserve. Adding to the fuel cost, labor, interest and depreciation, the cost per ton should not be over $0.30. In the above system there will be plenty of distilled water available for filling cores. Distilled water ice can be made at a slightly increased cost.

The author concludes with an interesting description of an exhaust steam absorption ice plant installed in the South, in which there are two 2,500-kilowatt turbines running condensing, also a 75-horsepower turbine direct connected to centrifugal pump. The guaranteed steam consumption of the turbine was 70 pounds per horsepower-hour when operating with 200 pounds steam and exhausting against 25 pounds back pressure, this being 20 pounds more than when exhausting into the feed-water heater at atmospheric pressure. In this plant the average cost of producing ice during the entire year was $0.4062 per ton.

TABLE 2—COST OF MANUFACTURING RAW-WATER ICE—100 TONS CAPACITY—VARIOUS SOURCES OF POWER

(From General Electric Review, July, 1914)

	Motive Power	Using	h.p.	Total h.p.-Per hr. Day	Fuel per h.p.	Fuel or Power Per Day	Unit Cost of Fuel	Fuel Cost per Day	Tons of Ice per Unit of Fuel	SUNDRY COSTS			LABOR COST					Total Cost per Day	Cost per h.p. of Power Plant
										Oil Waste, Etc.	Ammonia	Calcium	1 Day Engineer	1 Night Engineer	4 Ice Pullers	1 Extra Man	2 Firemen		
1	Simple Steam Engine	Bituminous Coal	367 i.h.p.	8808	3½ lb.	15.4 tons	$2.00/ton 2.50/ton 3.00/ton	$30.80 38.50 46.00	6.38 Tons Ice per Ton of Coal	2.00	2.50	0.50	5.00	3.25	8.00	2.00	5.00	$59.05 66.75 74.45	$32.00
2	Compound Condensing Engine	Bituminous Coal	375 i.h.p.	8952	2 lb.	8.95 tons	2.00/ton 2.50/ton 3.00/ton	17.90 22.38 26.85	11.15	3.50	2.50	0.50	5.00	3.25	8.00	2.00	5.00	46.15 50.63 55.10	40.00
3	Producer Gas Engine	Bituminous Coal	360 b.h.p.	8640	1¼ lb	5.4 tons	2.00/ton 2.50/ton 3.00/ton	10.80 13.50 16.20	18.5	3.50	2.50	0.50	6.00	4.00	8.00	2.00	4.00	*41.30 44.00 46.70	65.00
4	Producer Gas Engine	Anthracite Coal	360 b.h.p.	8640	1 lb.	4.31 tons	3.00/ton 3.50/ton 4.00/ton 4.50/ton 5.00/ton 5.50/ton 6.00/ton	12.93 15.09 17.24 19.40 21.55 23.71 25.86	23.2	3.50	2.50	0.50	6.00	4.00	8.00	2.00	4.00	43.43 45.59 47.74 49.90 52.05 54.21 56.36	65.00
5	Producer Gas Engine	Lignite	360 b.h.p.	8640	1¾ lb	7.56 tons	2.00/ton 2.50/ton 3.00/ton	15.12 18.90 22.68	13.2	3.50	2.50	0.50	6.00	4.00	8.00	2.00	4.00	45.62 49.20 53.18	65.00

TABLE 2—COST OF MANUFACTURING RAW-WATER ICE—100 TONS CAPACITY—VARIOUS SOURCES OF POWER

(From General Electric Review, July, 1914)

										SUNDRY COSTS			LABOR COST						
	Motive Power	Using	h.p.	Total h.p.-Per hr. Day	Fuel per h.p.	Fuel or Power Per Day	Unit Cost of Fuel	Fuel Cost per Day	Tons of Ice per Unit of Fuel	Oil Waste, Etc.	Am-monia	Cal-cium	1 Day Engi-neer	1 Night Engi-neer	4 Ice Pul-lers	1 Extra Man	2 Fire-men	Total Cost per Day	Cost per h.p. of Power Plant
6	Gas Engine	Natural Gas 1050 B.t.u. Value	360 b.h.p.	8640	10 cu. ft.	86400 cu. ft.	$0.20 per M 0.25 per M 0.30 per M 0.35 per M	$17.28 21.60 25.92 30.24	864 cu. ft. gas req'd per Ton of Ice produced	3.50	2.50	0.50	6.00	4.00	8.00	2.00		$43.78 48.10 52.42 56.74	$35.00
7	Oil Engine	Fuel Oil	360 b.h.p.	8640	0.075 gal.	648 gal.	0.02½ gal. 0.03 gal. 0.03½ gal. 0.04 gal.	16.20 19.44 22.68 25.92	0.155 Tons Ice per Gal. of Oil	3.50	2.50	0.50	6.00	4.00	8.00	2.00	4.00	46.70 49.94 53.18 56.42	40.00 to 60.00
8	Motor	Elec-tricity	327 on wire	7848	0.746 kw.	5854 kw.	0.00½ kw. 0.00⅝ kw. 0.00¾ kw. 0.00⅞ kw. 0.01 kw.	29.27 36.48 43.90 51.23 58.54	58.54 kw-hr req'd per Ton of Ice produced	2.00	2.50	0.50	5.00	3.25	8.00	2.00		47.51 59.63 67.15 74.48 †81.79	16.00

Water estimated at 70 deg. Fahr. Suction pressure 15 lb. Power included for pumping water with 60 foot head. *Minimum. Condenser pressure 185 lb. Ammonia used as refrigerant. †Maximum. Above costs do not include interest on investment, depreciation, taxes, insurance, management or any other fixed charges. Power is based on summer conditions. Less power will be required if yearly average is figured on. Oil engine fuel requirements based on engines of Diesel type.

It is further pointed out that while it is perhaps advisable to use the exhaust from the auxiliaries in condensing plants, in all cases it is entirely possible to bleed sufficient steam from an intermediate stage of a condensing turbine.

A 2,500-kilowatt turbine running at full load with 27½ inches of vacuum used 14.85 pounds steam per kilowatt-hour. When the turbine was operated under the same conditions, except that 5,000 pounds of steam at from 2 to 5 pounds pressure was extracted per hour, the steam consumption was increased to 16.32 pounds. This is an increase of 3,675 pounds per hour, and as the 5,000 pounds of steam will make 85 tons of ice, the extra steam amounts to 43 pounds per hour per ton of ice. If the engine is operated at half load, and 5,000 pounds of steam is extracted per hour, the steam consumption will be 16.8 pounds, in which case the extra steam will amount to 28½ pounds per hour per ton of ice. It is therefore more economical to extract steam from a condensing turbine operating at half load than at full load, but in neither case is the economy as high as when the exhaust from non-condensing auxiliaries is employed.

About the same amount of extra steam would be required to make 85 tons of ice if the turbine runs at full capacity, condensing, and current is taken from it to drive electrically-driven compression machines as for an absorption machine of the same capacity operating on steam bled from the turbine. In the first case, however, the available output of the turbine would be reduced by the amount of the current consumed by the compression machine, while in the second its full capacity would be available for commercial purposes.

In the discussion which followed the reading of this paper it was stated that about a third of a million tons of ice, or about 10 per cent of Chicago's total requirement, is now being made in eighteen plants driven by central station power. On account of the low increment cost of taking on load of this kind, power is sold for ice making at 1 cent per kilowatt-hour.

CONDENSATION OF GASOLINE FROM NATURAL GAS

THE CONDENSATION OF GASOLINE FROM NATURAL GAS. George A. Burrell, Frank M. Seibert and G. G. Oberfell. Bulletin 88, Petroleum Technology 20, United States Department of the Interior, Bureau of Mines, February, 1915. 100 pp., 18 figs.

A comprehensive review of the development of methods first put into commercial use in about 1904 for the manufacture of gasoline from the various gases and entrained vapors escaping from combined oil and gas wells. "Petroleum in the so-called paraffin oil fuels consists of hydrocarbons of the paraffin series, which range from the heaviest oil to the lightest gas." Chemically the more important of the series are the following:

	Boiling Point		Liquefying at		
	° C.	(° Fahr.)	° C.	(° Fahr.)	
Methane (CH_4)...	—160	(—256)	—95.5	(—139)	and 735 lbs. pressure
Ethane (C_2H_6)....	— 93	(—135)	—81.8	(—115)	" 807 " "
Propane (C_3H_8)...	— 45	(— 49)	35	(95)	" 664 " "
Betane (C_4H_{10})....	1.0	(33.8)	97	(206)	" 647 " "
Pentane (C_5H_{12})...	36.4	(97)			
Hexane (C_6H_{14})...	68.9	(156)			
Heptane (C_7H_{16})..	98.4	(209)			

The first four of these hydrocarbons, as will be seen from their boiling points, are gases under ordinary conditions, the last three liquids, the latter group being the chief constituents of that very definite fuel—gasoline. In the earth the gases may be free or in solution in the liquid constituents depending largely on pressures. When the pressure is relieved these gases escape from the solution, producing, in expanding, a certain amount of refrigeration which may precipitate some of the heavier constituents, such as paraffin.

The process by which the less volatile of the gaseous constituents is recovered in the form of a "condensate," and blended with naphthas to form commercial gasoline, is one of refrigeration, though in the majority of cases refrigerating machines proper are not employed. The gases are compressed ine one, two or three-stage compressors and cooled. Although products of widely different gravities and boiling points are separated out in different parts of the cooling apparatus, it appears that they are again mixed, before being blended, with the lower gravity oils. The more volatile constituents of the condensate, which might give rise to dangerous pressures in shipping containers, are allowed to escape by a process of "weathering."

The complexity of problems having to do with the liquefaction of mixtures of various gases having different boiling points occurring in varying and usually unknown proportions, under which conditions the same pressure and temperature may, but in the general case will not, liquefy the same gas, evidently limits research principally to "cut and try" methods. Pressures such as will give the best results under existing commercial conditions and temperatures of cooling media available are employed. Blending and weathering is carried out principally for the purpose of rendering the products of condensation safe to transport, those constituents not being readily utilizable in the form of gasoline being returned to the mother gas, which may be used for heating, lighting or pumping purposes, or, as seems to be the more general method, are allowed to go to waste. The economic crime of the latter method is evident from the statement of the bulletin, that "unless greatly contaminated with air it is superior to that ordinarily used in cities and towns."

The danger from explosions of the explosive mixtures that may be formed by the volatile oils and gases mixed with air drawn into the systems operating under a vacuum is apparent to refrigerating engineers, especially when it is remembered that the pressures employed may range, according to the nature of the gas, from as low as 50 pounds to as high as 350 pounds per square inch.

Aside from the danger of explosion the infiltration of air has the effect of reducing the condensate yield through dilution of the gases, which prevents them from being liquefied at the usual pressures. As oxygen is not found as a constituent of natural gas, its presence in the system is an absolute indication of air leaks.

That mechanical refrigeration will play an important part in the future of this enterprise seems to be assured by the present success of such systems, but the use of low temperature refrigeration will find its maximum application only after a market has been developed for the more volatile constituents of natural gas, which are now so largely wasted or at best utilized in only the gaseous form.

The value of gasoline that has been wasted in the past amounts in some instances to as high as 20 per cent of the oil produced, and 10 per cent is said to be a conservative estimate of the loss under average conditions.

REFRIGERATION IN TREATMENT OF TYPHOID FEVER

USE OF REFRIGERATION IN THE TREATMENT OF TYPHOID FEVER. *Le Froid,* Vol. II, No. 8. August-December, 1914. (Appeared June, 1915.) 4 pp.

It is interesting to note that the use of refrigeration in the treatment of typhoid fever is becoming more and more general.

On January 5, 1915, Dr. Franz Glénard lectured before the Académie de Médicine on the beneficial effects of cold baths in the treatment of typhoid fever, according to a method originated by Brandt and introduced in France 40 years ago. He stated that the natural mortality from this contagious disease was 20 per cent, increasing after epidemics to as high as 60 per cent (siege of Paris, 1870), while by the use of cold baths the loss was kept down to 12 or 10 per cent. The latter are nearly twice as effective as are mixed treatments in which some anti-thermics, such as salycilic acid or quinine is administered beside the baths.

The method of procedure adopted in ordinary cases of typhoid fever is as follows: Whenever the rectal temperature, which must be taken every 3 hours, day and night, from beginning to end of sickness, reaches or exceeds

39° C. (102.2° F.), the patient is to take a bath for 15 minutes in water cooled to 68° F. At start, middle and end of bath, head and neck to be splashed for one minute with water at 59° F. During intervals between the first baths apply cold compresses to alleviate the local symptoms not removed by the baths. Light nourishment to be given after each bath. Give frequent drinks of cold water, in severe cases half to 1 quart good wine in 24 hours.

In irregular or complicated forms of the disease another method is resorted to. The initial temperature of the bath is reduced from 97° to 68° F. within 20 or 30 minutes. If the refrigerating action appears to predominate over the stimulating action, a half-bath of 82° F. for 5 to 10 minutes with very cold showers is recommended to induce the stimulating action, after which the regular method is followed again until the complications have disappeared.

After giving further detailed suggestions the author concludes that hospitals for the treatment of such cases should have rooms cooled to less than 68° F., with bathing facilities in the center. Very much can be accomplished also by means of ice used in bags, for which the French paper gives useful hints.

BOILER LOSSES

A Study of Boiler Losses. By A. P. Kratz, Bulletin No. 78, Engineering Experiment Station, University of Illinois, April, 1915. 72 pp., 31 cuts, 19 tables.

The purposes of the investigation, as outlined by the author, were to determine the "conditions prerequisite for the continuous operation of the new power plant at the University of Illinois, and to permit a detailed study of the boiler and furnace losses under varying conditions of load, depth of fuel bed and draught. A majority of the trials were made with fresh coal from Danville. One series, however, was made with weathered coals from Sangamon, Williamson and Vermilion counties to determine the effect of weathering. The plant consisted of Babcock & Wilcox boilers equipped with stokers of the chain-grtate type. The results of the 24 tests are shown in 16 tables, in which the items are arranged in accordance with the Boiler Test Code of the American Society of Mechanical Engineers. Three efficiencies were determined, viz.: the efficiency of furnace and grate, the efficiency of the furnace, and the efficiency of the boiler, exclusive of boiler and grate. The results of the tests indicate that the best efficiency under full-load conditions was obtained with a fuel bed of from 7 to 7½ inches in thickness. Under normal conditions of load and fire the over-all efficiency averaged about 65 per cent. This represented actual operating efficiency, and as such was in accord with good practice. The efficiency of the grate and furnace was fairly constant for all loads and depths of fuel bed. The loss in efficiency at higher loads was due to reduced heat absorption by the

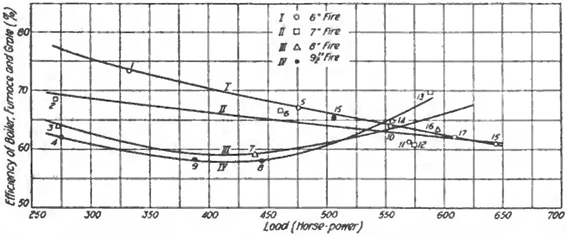


Fig. 1.—Relation between load and efficiency of boiler, furnace and grate.

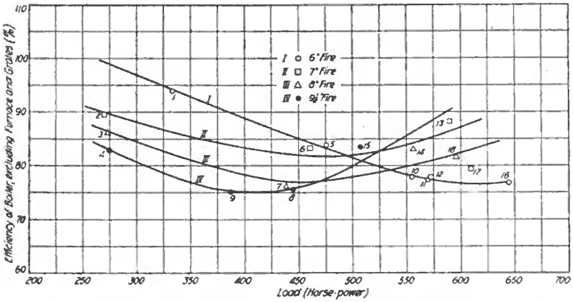


Fig. 2.—Relation between load and efficiency of boiler, excluding furnace and grates.

boiler. The curves shown in Fig. 1 and Fig. 2 allow interesting comparisons to be made between the efficiency of the boiler, furnace and grate, and the boiler only, at different loads and different thicknesses of fire beds. The deterioration due to weathering for six years is shown in Table 1.

Table I.—Deteriorating Due to Weathering

Coal.	How Stored	Jan. 1908.		Jan. 1914.	
		Per Cent Passing a ¼" Screen.	B. t. u. per Lb. Dry Coal 7 Days After Mining.	Per Cent Passing a ¼" Screen.	B. t. u. per Lb. Dry Coal at Time of Test.
Sangamon Co. Nut...........	Open bin	1.2	11800	31.9	11179
Sangamon Co. Screenings.....	Covered bin	26.3	11684	45.1	10497
Williamson Co. Nut..........	Covered bin	1.4	12341	13.9	12103
Williamson Co. Screenings....	Covered bin	38.8	12287	42.9	12153
Vermilion Co.. Nut..........	Open bin	13.6	12412	35.0	11985

The paper closes with a valuable and instructive appendix of eighteen pages containing formulas and forms for calculating boiler trials.

KILLING OF PLANT TISSUE BY LOW TEMPERATURE

The Killing of Plant Tissue by Low Temperature. W. H. Chandler. Missouri Sta. Research Bul. No. 8. pp. 143-309.

The work reported in this paper was begun during the season of 1904-5. While the studies were undertaken primarly to determine the effect of certain cultural methods on the hardiness of peach fruit buds under climatic conditions that prevail in the southern half of Missouri, they have been so extended as to embrace a general study of the "freezing to death" of plant tissue. In addition to peaches and other orchard fruits many vegetables and other cultivated plants have been included in the author's experimental studies. The results of this work are tabulated and fully discussed in connection with the results secured by various investigators. The subject matter is presented under the following general headings: Review of literature on freezing, effect of sap density on temperature, other features that influence the freezing to death of plants, effect of previous exposure to temperature slightly above killing temperature, relation of low temperature to peach growing, varieties with the longest rest periods, effect of vigor of threes on rest periods, breeding varieties hardy under Missouri conditions, killing of apples and killing of cherries and plums.

The most commonly accepted theory dealing with the freezing to death of plant tissues seems to be that killing from cold is due to the withdrawal of water from the protoplasm. Results of many investigations show that the water generally moves out of the cells to form ice crystals in the intercellular spaces. The amount of water loss necessary to result in death varies with different plants and different tissues. Experimental data secured by the author lead him to conclude that freezing to death cannot be attributed to precipitation of proteids, as claimed by some investigators.

The author's experiments with seedlings of various plants and with twigs bearing small apples, peaches and cherries show that for plant tissues that kill at relatively high temperature the killing temperature is reduced whenever the sap density or molar concentration of the sap of the tissue is increased. In these studies increases in sap density were brought about by using mineral salt solutions and also by watering the seedlings sparingly. When sap density was reduced by shading the plants the amount of killed tissues at a given temperature was increased.

Practically all plants used in freezing experiments in this work were tested with reference to the effect of rapid thawing. The results show that in addition to ripe apples, pears and the leaves of *Agave americana* observed by Muller-Thurgau and Molisch, leaves of lettuce kill at slightly lower temperature if they are thawed slowly than if thawed rapidly. In the case of all other tissues tested, either by the author or by others, however, including unripe apples and pears, there is no indication that the rate of thawing has anything to do with the amount of killing at a given temperature. Rapid wilting of tissue has not generally increased the resistance of plants to low temperature over that of unwilted tissue with a dry surface. Tissue with a wet surface killed worse at a given temperature than did tissue with no moisture on the surface. Slow wilting or partial withholding of water through a long period was found to increase the resistance of tissue to low temperature. In case of hardy winter buds and wood a rapid decline in temperature greatly increased the severity of injury from a given low temperature. Previous exposure of plant tissue to low temperature above that at which the tissue kills seems to increase its resistance to low temperature.

As between different plants there appears to be no constant relation between the rate of growth of plant tissue and resistance to low temperature.

EFFECT OF TEMPERATURES ON POTATO SCAB ORGANISM

EFFECT OF TEMPERATURES ON GERMINATION AND GROWTH OF THE COMMON POTATO SCAB ORGANISM. Michael Shapovalov, Assistant Plant Pathologist, Maine Agricultural Station. *Journal of Agricultural Research,* May, 1915. 5 pp.

The article describes the results of experiments conducted by the writer on several strongly pathological strains isolated from diseased specimens of potatoes received from Maine, Vermont and Wisconsin. Slides of the cultures placed in incubators of the required temperatures were removed hourly for microscopic examination.

The maximum temperature for growth was found to be a little below 41° C. (106° F.), the optimum from 25° C. (77° F.) to 30° C. (86° F.), and the minimum about 5° C. (41° F.).

Germination was most rapid between 35° C. (95° F.) and 40° C. (104° F.); but these temperatures were found unfavorable for long-continued growth. Minimum temperatures of germination were not determined at this time, but the writer's previous experiments show that is somewhere near 5° C. (41° F.).

These determinations, which indicate that the minimum temperature for both germination and growth is about 5° C. (41° F.), are a valuable supplement pathologically to the results of potato storage experiments made in the New Hampshire Experiment Station, mentioned in the May A. S. R. E. JOURNAL, under the head of Investigations Into the Preservation of Food Products, the results of which indicated that potatoes will keep for two hundred days without sprouting or loss of weight when stored in still air at from 38 to 40° F.

Müller-Thurgau found that in the common potato the accumulation of sugar practically ceases at 8° C. (46.5° F.).

It would seem reasonable to conclude, therefore, that lower temperatures than 38 to 40° F. are unnecessary, not only for the prevention of sprouting and loss of weight by evaporation, but also for the prevention of rot, due to the development of the potato scab organism.

NOTE:—It is interesting to note that the results obtained by the New Hampshire station in its investigation of the behavior of the common potato under refrigeration are diametrically opposite to those in the *Journal of Agricultural Research* for sweet potatoes. All of the experiments showed that the disappearance of starch and the accumulation of sugar takes place more rapidly and proceeds to a greater extent at low than at high temperatures. It is not rapid between 0 and 4° C. (32 and 39° F.). The experiments were of short duration, because the sweet potatoes rotted after about six weeks under refrigeration. The great difference in the keeping properties of common and sweet potatoes is probably due to the greater sugar content of the latter. How to carry this crop successfully under refrigeration will undoubtedly prove a formidable problem for the plant pathologist to solve.—*Reviewer.*

FREEZING POINT OF MILK

THE FREEZING POINT OF MILK. J. B. Henderson and L. A. Meston. *Chem. News,* Nos. 2870, pp. 259-261; 2871, p. 275; 2872, pp. 283, 284.

It is reported that results from 63 samples of milks obtained during the past five years show that in 54 the added water content indicated by the freezing point was from 0.2 to 9.9 per cent higher than that calculated on an 8.5 per cent solids-not-fat basis, the average being 4 per cent. In five cases the calculation of added water by the two methods was practically identical. In three other cases the estimated added water on the 8.5 per cent solids-not-fat basis from 1.7 to 3.9 per cent lower than by the freezing point.

LIQUID AIR AS AN EXPLOSIVE

VERWENDUNG VON FLÜSSIGER LUFT ALS SPRENGMITTEL NACH EINEM NEUEREN VERFAHREN. M. Przyborski. *Zeits. für Eis und Kälte-Industrie,* Vol. VII, No. 9, March, 1915. 2 pp.

Data are given on the use of liquid air as an explosive, in particular on the Kowatsch system.

Professor Linde appears to have been the first to introduce successfully an explosive consisting of ground charcoal and liquid air. Since then Claude and d'Arsonval, in France, have further developed these processes, and finally Kowatsch and Baldus, in Germany, have lately worked it out still more fully, partly under the pressure of the present demand for an explosive of exclusively German manufacture.

In order to avoid excessively rapid evaporation of the liquid air, Kowatsch, who uses a cardboard cartridge, introduces first a cartridge with dry carbon into the drill hole separately and without any liquid air, which is put in only just previous to ignition. This process permits reducing to a minimum the period of evaporation, and also increases the factor of safety of operation.

The cardboard cartridge contains a perforated pipe, in which there is a mixture of infusorial earth with oil and asphaltum or lampblack and paraffin, neither of which is explosive in itself. This center pipe contains another cardboard tube over which a third cardboard tube is set, serving as an exhaust pipe for the products of evaporation of liquid air. If several drill holes have to be exploded simultaneously, the electric connections are made accordingly.

The liquid air which has to be used for each hole is kept in a precisely determined quantity in a small bottle, the opening of which is provided with a metal tube and conical nozzle, connected with the central tube of the cartridge. To load the cartridge all that is necessary is to lift the rear end of the bottle. The liquid air is raised by the pressure of its own products of evaporation and gradually passes into the cartridge. At the instant when the charge is ignited by the electric spark, the liquid air combines with the charge and there follows an explosion of exceptional violence.

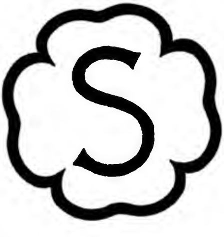
Among the advantages of this method of explosion is the fact that, in the first instance, the materials used are not explosive in themselves, and the explosive mixture is fotmed only at the very last moment in the bore hole itself. In the second place, should an explosion be missed, the liquid air will evaporate, and the resmaining cartridge is perfectly harmless.

A. S. M. E. BOILER CODE

At a recent meeting of the Boiler Code Committee of the American Society of Mechanical Engineers, held in Buffalo, June 22 to 25, the following resolution was offered to the Council:

> It is the opinion of the committee that the official symbol or stamp is to be used to indicate that the American Society of Mechanical Engineers' rules have been complied with in the construction of the boiler. The stamp shall be affixed by the manufacturer. Certification may be governed by law or contract.

The resolution was adopted by the Council, and the stamp appearing below, to be used as prescribed in the code, will be open to general use.



The work of the Code Committee in formulating the code, which was reviewed in the March number of The Journal, is a long step in advance. The code itself is sufficient evidence that the right men were entrusted with the duty, and the promptness with which it is being adopted shows conclusively that need for such a code was imperative. After recommendation by the executive committee of the Master Boiler Makers' Association, the use of the Boiler Code as a set of rules for stationary boiler construction was fully endorsed by the association at its ninth annual convention at Chicago. The code has also been adopted by the Industrial Board of the State of Pennsylvania, as well as by the Board of Boiler Rules of the City of Detroit. It is said to be more than likely that it will be the standard of boiler construction in a large proportion of the States before the end of the year.

A very important question, that of interpretation of the Boiler Code rules, has arisen. It has already been met, however, by the Council in adopting the following resolution offered by the Code Committee:

> Your committee requests that it be empowered to make rulings where inquiries are made regarding construction not covered by the code and to interpret any parts of the code.

In order that no obstacle be placed in the way of the application of the code by the industry, arrangements were made for quarterly meetings of the committee.

The work of the committee was further extended by the adoption by the Council of a third resolution, which is as follows:

> Your committee requests that it be empowered to take up the subject of (1) economizers; (2) pressure vessels; (3) rules for operation and care of steam boilers and pressure vessels, and (4) recommendations.

EXTRACTION OF HUMIDITY FROM BUILDINGS BY REFRIGERATION

Extraction of Humidity from Newly-Constructed Buildings by Means of Refrigeration. M. A. Kappen. Bulletin of July to December, 1914, of the Société des Ingenieurs Civils de France.

In this article, which is of more than usual interest on account of the insight it gives into details of masonry construction employed in damp climates, the writer refers to previous articles describing means of preventing the water from rising in excess quantities by capillarity in stone walls, means of ventilating these walls so that they are freed from water absorbed during rainstorms, and finally describes methods by which newly-constructed masonry buildings may be rendered habitable within a minimum space of time by the use of refrigeration.

The various sources of moisture in new dwellings arise from four different causes manifesting themselves in the following chronological order: First, original humidity in building materials, such as mortar, plaster, etc.; second, moisture picked up by capillarity from the soil; third, atmospheric humidity condensed on the cold materials; fourth, water of infiltration, rain, etc., during construction.

Past methods for allowing buildings to be occupied promptly after completion have involved the use of heat generated with or without the production of carbonic acid in forced air circulation systems. Methods formerly employed for the protection of old buildings against the ravages of moisture entering by capillarity and condensation are far too slow to meet present-day requirements for quickly removing the original humidity of construction so that the buildings may be occupied at the earliest possible moment after completion of plastering. Furthermore, the attempt to drive off all humidity from fresh mason materials involves a physical error. Capillarity is stronger on the colder side of the wall. In warming the air of an enclosure by the various heating systems the walls are warmed by convection, and the humidity not evaporated by the hot air, which soon becomes too fully saturated to absorb it, is driven back, or rather is drawn back, by capillarity into the interior of the wall. Where carbonic acid is liberated by heaters the currents of air and carbonic acid brought in contact with the surface of the plaster renders it almost completely impermeable, preventing the escape of the humidity when it again returns, due to the equalization of temperatures. These methods of drying by means of heat produce then an apparent dryness, but leave sufficient moisture in the walls treated, so that when it returns to the surface the plaster may be detached, the paint destroyed, and the walls rendered moist and unsanitary.

What heat is incapable of accomplishing in the way of removing all of the humidity, so that new buildings can be made habitable immediately after completion, may be readily effected by the use of refrigeration. Only by the judicious use of low temperatures is it possible to dry out new buildings to the point of habitability in a few hours. The most favorable temperature for this work is 4° C. (39° F.). At certain seasons of the year, when the temperature is about 4° C., certain curious modifications in the materials of walls frequently take place. Following rains or periods of unusual humidity certain constituents of the mortar are dissolved by moisure entering the wall by capillarity. When the humidity decreases these constituents are carried to the exterior surface, where they are deposited on the evaporation of the moisture, forming a white discoloration. This effect is most likely to happen when the mortar is rich. The deposit, consisting of sulfate of lime and other salts, is washed off.

Assuming a hygrometric condition of 80° in the air of the room in which the walls are freshly plastered with mortar, only 3½ to 4° depression of the temperature is necessary in order to arrive at the dew point and to precipitate the atmospheric humidity of the air cooled. In other words, at 80° saturation and 7½° C. (45.5° F.) it is only necessary that the humid air be reduced in temperature to 4° C. to cause precipitation. In order to get more effective results the temperature might be reduced to 0° C., but this might damage the more delicate plasters in which slight freezing might take place. The final coat of plaster, which is less porous, can be applied after the water has been extracted from the more porous under-coats. Drying the preceding coats of plaster would hasten the completion of the total drying, but might possibly result in a less perfect adherence of the next coat.

It is claimed for the refrigeration method that it will withdraw the moisture from walls without closing the pores or cracking or crazing the surface. By this method the wall escapes the harmful effects due to the shrinking of motor under the application of heat.

In practice aluminum vessels containing the cooling medium are suspended on frames in such a manner that they can be raised or lowered and secured at any required height between the floor and the ceiling, where they are left in position for a sufficient length of time for the extraction of the moisture.

In conclusion, the writer eulogizes refrigeration: "Cold, that potent factor of modern welfare, now produced everywhere almost as easily as heat, has found a most important application, which, in time, will become a powerful co-laborator of hygienists in making new buildings sanitary. A little cold passed over the surfaces of the walls now permits new houses to be occupied not only immediately but safely."

SOCIETY AFFAIRS

ADVANCE PROGRAM

OF

FOURTH WESTERN MEETING

The Fourth Western Meeting of The American Society of Refrigerating Engineers will be held in San Francisco, Cal., on September 23 and 24, 1915. In this, and the last two issues of the A. S. R. E. JOURNAL, is the itinerary of a special train movement on account of our meeting in San Francisco. The special train will leave Chicago, Ill., on Wednesday, September 15, at 6:10 o'clock P. M., and will arrive in San Francisco on Tuesday evening, September 21, at 6:30 P. M.

The headquarters of the Society will be at the Inside Inn, located within the grounds of the Panama-Pacific International Exposition. The rates for rooms, European plan, include unlimited admission to the Exposition grounds during stay at the hotel. All rooms are outside rooms, with sunlight during some portion of the day, and the hotel accommodations are first-class in every respect. The rates are as follows:

Rooms without bath, one person in room, $2.00 per day; two or three persons in one room, $1.50 per person per day.

Rooms with bath, one person in room, $3.00 to $4.00 per day, according to location; two persons in one room, $2.00 to $2.50 per person per day, according to location.

The sessions of our meeting, as well as the International Engineering Congress, will be held in the Exposition Memorial Auditorium at the Civic Center in San Francisco.

The entertainment programme and list of papers to be read and topics discussed, subject to change, are as follows:

Tuesday, September 21:

Arrive in San Francisco at 5:30 P. M. Dinner at 8 P. M. at Portola-Louvre Café, $2.50 per plate.

Wednesday, September 22:

Morning—Sightseeing trip around San Francisco, viewing Golden Gate Park, Cliff House and Presidio, and returning by the Exposition grounds. Price, $1.00 per person. Time, 2½ hours. Luncheon at the Portola-Louvre Café, 50 cents per person.

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Afternoon—Sightseeing trip from San Francisco via Creek route ferry to Oakland, Berkeley, Claremont, Piedmont and Lake Merritt to Canyon Inn, Hayward, and returning via Foothill Boulevard. Price, including dinner at Canyon Inn, $3.50 per person.

Thursday, September 23:

Morning—Meeting of A. S. R. E.

Roll call.

Reading of minutes of Tenth Annual Meeting.

Report of Tellers of Election of Members.

Paper on "Oil Engine Driven Raw Water Ice Manufacturing Plant." President Louis K. Doelling, New York, N. Y.

Discussion of topic, "Raw Water Ice Making Systems."

Afternoon—Attend session of International Engineering Congress, when papers on refrigeration will be presented.

Friday, September 24:

Morning—Meeting of A. S. R. E.

Paper on "Fruit Precooling and Precooling Plants."

Paper on "The Rate of Heat Transfer in Double Pipe Brine Coolers." Van R. H. Greene, New York, N. Y.

Discussion of topic, "Power Required to Agitate Brine in Can Ice Tanks."

Afternoon—A. S. R. E. Day at Exposition. Dinner at 7 P. M. at Inside Inn, $2.00 per person.

Saturday, September 25:

Afternoon—Steamboat excursion around the Bay of San Francisco, steamer leaving San Francisco at 2 P. M., and returning at 6 P. M. All principal points of interest will be visited, including a trip through the Golden Gate. Price, $1 per person, providing a party of 100 can be made up.

Evening—Banquet at the Hof Brau Café, $5.00 per cover, including wine.

SECOND PAN-AMERICAN SCIENTIFIC CONGRESS

The Second Pan-American Scientific Congress will meet in Washington, D. C., from December 27, 1915, to January 8, 1916, inclusive, under the auspices of the Government of the United States. The First Congress was held in Santiago, Chile, in December, 1908.

The organization and procedure of the Second Congress will be in charge of an executive committee of the following, with power to enlarge itself:

William Phillips, Third Assistant Secretary of State, Chairman *ex officio*.
James Brown Scott, Secretary, Carnegie Endowment for International Peace, Vice-Chairman.
William H. Welch, President, National Academy of Sciences, Honorary Vice-Chairman.
John Barrett, Director-General, Pan-American Union.
W. H. Bixby, Brigadier General, U. S. A., retired.
Philander P. Claxton, Commissioner of Education.
William C. Gorgas, Surgeon General, U. S. A.
William H. Holmes, Head Curator, Smithsonian Institution.
Hennen Jennings, former President, London Institution of Mining and Metallurgy.
George M. Rommel, Chief, Animal Husbandry Division, Bureau of Animal Industry, Department of Agriculture.
L. S. Rowe, President, American Academy of Political and Social Science.
Robert S. Woodward, President, Carnegie Institution of Washington.

The organization officers are John Barrett, Secretary-General, and Glen Levin Swiggett, Assistant Secretary.

The following are the sections into which the Congress will be divided, along with the name of the chairman of each section:

I. Anthropology—Dr. Holmes.
II. Astronomy, Meteorology and Seismology—Dr. Woodward.
III. Conservation of Natural Resources—Agriculture, Irrigation and Forestry—Mr. Rommel.
IV. Education—Dr. Claxton.
V. Engineering—General Bixby.
VI. International Law, Public Law and Jurisprudence—Dr. Scott.
VII. Mining and Metallurgy, Economic Geology and Applied Chemistry—Mr. Jennings.
VIII. Public Health and Medical Science—General Gorgas.
IX. Transportation, Commerce, Finance and Taxation—Dr. Rowe.

The sessions of the Congress will be held in the Pan-American Union Building.

On invitation of the Secretary of State of the United States, as well as of John Barrett, Secretary-General of the Congress, that The American Society of Refrigerating Engineers send a delegate to this Congress, President Louis K. Doelling has appointed N. H. Hiller, Carbondale, Pa., to represent the Society.

NON-MEMBERS MAY SUBSCRIBE TO A. S. R. E. JOURNAL

At the last annual meeting of the Society a motion prevailed authorizing the acceptance of subscriptions to the A. S. R. E. JOURNAL from non-members, the subscription price being left to the Council to decide. The Council has set the annual subscription price at the small amount of $3.00, in order that the work of the Society may be of benefit and available to the industry generally. Of course, the price of a yearly subscription to the A. S. R. E. JOURNAL is included in the dues of each member.

The JOURNAL will be an important means of increasing our membership, for as soon as non-member subscribers become acquainted with and follow the work of the Society they will desire to further co-operate and then become members.

USE THE TITLE OF YOUR GRADE OF MEMBERSHIP

President Louis K. Doelling issued the following letter to the membership on June 28, and respectfully requests the co-operation of the membership along the line suggested:

"Permit me to suggest that wherever possible, on letterheads and in advertising matter, etc., members of the Society mention the fact that they are members of The American Society of Refrigerating Engineers. This will be of advantage to both you and the Society.

"The abbreviations of the titles of the various grades of membership, as given in paragraph B 30 of the By-Laws, are as follows:

"For Members—Mem. Am. Soc. R. E.
"For Associates—Assoc. Am. Soc. R. E.
"For Juniors—Jun. Am. Soc. R. E."

TRADE CATALOGS

(Below is given a list of trade catalogues received at the Society's office since the last issue of the A. S. R. E. JOURNAL.)

GIFFORD-WOOD COMPANY, Hudson, N. Y.—Bulletin No. 17. Pivoted bucket carrier for elevating and conveying coal, ashes, coke, stone, gravel, sand and other materials.

ARMSTRONG CORK COMPANY and ARMSTRONG CORK & INSULATION COMPANY, Pittsburgh, Pa.—Cork at San Francisco—The story of a unique exhibit of cork products and by-products at the Panama-Pacific International Exposition.

PHILADELPHIA PIPE BENDING COMPANY, Philadelphia, Pa.—Philadelphia ammonia condenser—atmospheric and double-pipe types.

THE 1915 YEAR BOOK

The Year Book for 1915 was mailed to the membership of the Society on June 12, and it is hoped each member received his copy. The Year Book is larger than previous editions, being 3¾" x 8", and contains 56 folios and cover. A geographical list of members has been added, in additional to the usual alphabetical list of members. Other matter contained in the book includes list of officers for 1915, list of past and present officers, committees, names of deceased members, depositories for the A. S. R. E. JOURNAL, calendar of past meetings and the constitution and by-laws.

PERSONAL

CHARLES L. CASE is general manager of The Norcross Brothers Company, general contractors, Worcester, Mass.

GEORGE BERNA is in Barrauquilla, Colombia, S. A., installing an ice manufacturing plant.

F. M. ADAY, former manager of the Boston office of the De La Vergne Machine Company, has been appointed manager of the Philadelphia office of the De La Vergne Company.

J. I. LYLE is Treasurer and General Manager of the Carrier Engineering Corporation, 39 Cortlandt Street, New York, N. Y., which corporation took over the business of the Carrier Air Conditioning Company.

INVITATIONS TO VISIT EXHIBITS AT SAN FRANCISCO

The following firms invite members of the Society to visit their exhibits at the Panama-Pacific International Exposition:

Armstrong Cork Company, Pittsburgh, Pa.
Foxboro Company, Foxboro, Mass.
National Tube Company, Pittsburgh, Pa.
York Manufacturing Company, York, Pa.

SPECIAL TRAIN TO SAN FRANCISCO MEETING

In order to insure a comfortable and pleasant journey for those of our members and friends who will attend the Fourth Western Meeting of The American Society of Refrigerating Engineers at San Francisco, Cal., September 23 and 24, as well as the International Engineering Congress, San Francisco, Cal., September 20 to 25, 1915, and the Panama-Pacific International Exposition, and inasmuch as a large number of those who will attend will either start from or pass through Chicago, Ill., or Kansas City, Mo., arrangements have been made with the Chicago, Burlington & Quincy Railroad Company to operate a special train de luxe from Chicago to San Francisco for these occasions.

The Burlington Route, Denver & Rio Grande and Western Pacific railroads have been selected as the route for this special train. This congenial way of traveling together will afford a delightful occasion to develop closer friendships and with environments rivaling club life. It is earnestly hoped that many members, with their families and friends, will avail themselves of this opportunity to attend our Fourth Western Meeting, the International Engineering Congress and the Panama-Pacific International Exposition, and to visit the West when its attractions are greatest and under most favorable conditions.

EQUIPMENT

The special train will be personally conducted over the entire journey from Chicago to San Francisco by a representative of the passenger department of the Burlington Route. It will be one of the handsomest trains ever assembled; the equipment will include a baggage car, in which is located a dynamo from which the train will be brilliantly electric-lighted, standard steel drawing-room sleeping cars, dining car and a compartment observation and library car. Every effort will be made to make this a most comfortable and memorable trip.

SCHEDULE

Leave—Chicago, Union Sta. (Canal and Adams Sts.)..	6:10 P. M.,	Sept. 15—Burlington Route
Arrive—Kansas City.......	8:00 A. M.,	Sept. 16—Burlington Route
Leave—Kansas City.......	9:50 A. M.,	Sept. 16—Burlington Route
Arrive—Denver	7:50 A. M.,	Sept. 17—Burlington Route
Leave—Denver	1:00 A. M.,	Sept. 18—Denver & Rio Grande R. R.
Arrive—Colorado Springs..	3:30 A. M.,	Sept. 18—Denver & Rio Grande R. R.
Leave—Colorado Springs...	4:30 A. M.,	Sept. 19—Denver & Rio Grande R. R.

Passing Through Royal Gorge at 7:45 A. M.

Arrive—Salt Lake City....	7:00 A. M.,	Sept. 20—Denver & Rio Grande R. R.
Leave—Salt Lake City.....	11:45 A. M.,	Sept. 20—Western Pacific Railway
Arrive—San Francisco....	6:30 P. M.,	Sept. 21—Western Pacific Railway

The departure from New York will be in special cars over the Pennsylvania Railroad, and the schedule will be as follows:

Leave—New York, Pennsylvania Station, 2:04 P. M., Sept. 14—Pennsylvania Railroad.

Leave—Philadelphia, Broad Street Station, 4:31 P. M., Sept. 14—Pennsylvania Railroad.

Leave—Harrisburg, Pennsylvania Station, 6:40 P. M., Sept. 14—Pennsylvania Railroad.

Leave—Pittsburgh, Pennsylvania Station, 1:15 A.M., Sept. 15—Pennsylvania Lines.
Arrive—Chicago, Union Station (Canal and Adams Streets), 5:00 P.M., Sept. 15—Pennsylvania Lines.

Members and their friends from the East who will not be able to avail themselves of the special service via the Pennsylvania Railroad, and also those from the South, will meet in the Union Station, Canal and Adams Street, Chicago, Ill., before the departure of our special train at 6:10 P.M. on September 15. Members and friends from points west of Chicago and Kansas City may board the special train en route.

RAILROAD FARES

There will be on sale at all ticket offices of all railroads round-trip tickets for this special train, with final return limit of three months from date of sale, but not to exceed December 31, 1915. Tickets will permit stop-overs at and all points west of Chicago and St. Louis within the time limit of the ticket, but must read via routes shown in schedule, namely, Burlington Route from Chicago or Kansas City to Denver, Denver & Rio Grande Railroad from Denver to Salt Lake City, and the Western Pacific Railway from Salt Lake City to San Francisco, as well as return route selected. Below are given the round-trip railroad fares from important points, but not including Pullman sleeping-car fares:

New York	$ 98.80	Pittsburgh	$ 81.25
Boston (via New York)	109.30	Columbus	74.18
Boston (via Albany and Buffalo)	104.20	Cincinnati	70.25
Philadelphia	95.20	Chicago	62.50
Harrisburg	91.05	Milwaukee	65.90
York, Pa.	92.15	St. Louis	57.50
Baltimore	92.95	Kansas City	50.00
		St. Paul	63.85

These rates apply returning via direct lines, which will include Los Angeles, San Diego, Grand Canyon, etc. To return via Portland, Seattle or Vancouver, the payment of $17.50 in addition to the above rates will be charged, and those returning via the North Pacific Coast may have their tickets routed via the Great Northern Pacific Steamship Company between San Francisco and Portland, with no extra charge for meals and berth while at sea. The time of travel of the palatial boats of the Great Northern Pacific Steamship Company between San Francisco and Portland about equals the fastest trains between these points, and this gives a delightful ocean voyage in connection with the trip.

Tickets are proportionately low from all points. The return trip from San Francisco is optional, but the entire routing must be decided upon at the time you purchase your ticket. A number of optional routes are given herewith. All tickets must be validated before leaving San Francisco, for which a charge of fifty cents is made.

PULLMAN FARES

The following Pullman sleeping-car fares will apply for the going trip on this special train, including all stop-overs scheduled from Chicago to San Francisco:

From Chicago	
Lower berth	$ 16.00
Upper berth	12.80
Compartment	45.00
Drawing room	56 00

From Kansas City	
Lower berth	$ 14.00
Upper berth	11.20
Compartment	39.50
Drawing room	49.00

In order that the necessary sleeping-car equipment may be reserved, it is earnestly requested that you make your reservations NOW. The amount of your Pullman fare from either Chicago, Kansas City or Denver to San Francisco should accompany your reservation, and be forwarded to W. H. Ross, Secretary, The American Society of Refrigerating Engineers, 154 Nassau Street, New York, N. Y. In the event you are not able to make the trip the money will be refunded.

All meals on dining cars, at hotels, etc., will be served à la carte at reasonable prices, and side trips and excursions at reduced rates.

BAGGAGE

One hundred and fifty pounds of baggage will be carried free, and by special arrangement members of the party may have access to the baggage car at any time, day or night. Suit cases and hand baggage may be taken in the sleeping cars.

TELEGRAMS

Telegrams for parties on board this special should be sent in care of The American Society of Refrigerating Engineers' Special Train.

WHAT TO DO IF YOU ARE GOING ON THIS SPECIAL TRAIN

1.—Notify W. H. Ross, Secretary, The American Society of Refrigerating Engineers, 154 Nassau Street, New York.

2.—Send to W. H. Ross, Secretary, The American Society of Refrigerating Engineers, 154 Nassau Street, New York, N. Y., the amount of Pullman sleeping-car fare from Chicago or Kansas City or Denver, the point of boarding special train, to San Francisco.

3.—Purchasing your railroad ticket from your local agent, which must read from your home city to Chicago or Kansas City or Denver, the point of boarding special train, to San Francisco, via the route and schedule of the special train. If you will return before December 31, 1915, your railroad ticket also must cover entire return routing.

4.—Do these now.

Itinerary

Wednesday, September 15.—Leave Chicago, Ill., Union Station, corner Canal and Adams Streets, 6:10 P.M., via Burlington Route. Dinner in dining car.

Thursday, September 16.—Arrive at Kansas City at 8:00 A. M. Breakfast will be served in the beautiful new Union Station, which has recently been completed at a cost of over six million dollars. The architecture of this building is said to be the finest specimen of its kind in the world. Leave Kansas City at 9:50 A. M. The day will be spent traversing the tremendously rich Missouri Valley and the rich farming section of Nebraska, the agricultural enterprises of which are among the most prosperous in the entire world. The picturesque Missouri River is crossed at Napier about 1 o'clock in the afternoon. During the night the train will climb the eastern foot-hill slope of the Rockies, which will be so gradual and unperceptibly accomplished that Denver, the mile-high metropolis of the Rockies and the gateway to the "Wonderland above the clouds," will be reached at 7:50 the following morning. Luncheon and dinner will be served in the dining car.

FRIDAY, SEPTEMBER 17.—Arrive at Denver at 7:50 A. M. Breakfast, luncheon and dinner will be served at the Albany Hotel, a few blocks from the depot, and which will be hotel headquarters during stop in Denver. Several large rooms, with bath, will be provided for our members without charge. During the morning the United States Mint, State Capitol, with its million-dollar museum, containing the finest geological collection in the West, and also many relics of the early frontier days, and many other points of interest in this "Queen City of the Plains," can be visited.

In the afternoon there will be a 52-mile valley, mountain and canyon trip to the summit of Lookout Mountain. The party will leave Denver in special parlor observation trolley cars, and the route passes through beautiful North Denver, "The Highlands," skirting Elitch's and Lakeside Parks to Clear Creek Valley, and following this veritable Eden to Golden. Just before entering Golden the route leads through the Clear Creek Placer Fields, where gold was discovered in 1859. At Golden a transfer will be made to high-powered automobiles for the one-hour climb up the mountain side over the new magnificent mountain highway to Lookout Mountain Park. A stop will be made at Wildcat Point—the summit of the mountain.

In the evening a theater party is planned for those who desire to attend. The train will leave Denver over the Denver & Rio Grande Railroad at 1:00 A. M., but sleeping cars will be ready for occupancy at the Union Station at 10:30 P.M.

SATURDAY, SEPTEMBER 18.—Arrive at Colorado Springs, Col., "City of Sunshine," at 3:30 A. M., where the train will be conveniently parked. Breakfast will be served at the famous Antlers Hotel at 6 o'clock, after which the party will take a carriage drive through the far-famed Garden of the Gods, passing through the wonderful wind-wrought obelisks, Balanced Rock, Mushroom Park, The Mesa, Glen Eye, etc., until Manitou, Col., that Swiss-like village, closely nestled at the base of Pike's Peak, famous for the curative properties of its nine soda, iron and sulphur springs, its sublime scenery and the crispness of its embracing air, is reached. Luncheon will be served at the Cliff House, Manitou. At 1:15 P. M. those who care to do so may take the Cog Wheel Route to the summit of Pike's Peak, 14,109 feet above sea level. The climb is made in eight and three-quarter miles, and stops are made at interesting points en route. The impressive view gained from the lofty altitude is one never to be forgotten, for nowhere in America is such a sight to be had, standing, as it were, on the backbone of the world. With the aid of the telescope in the observation tower objects may be clearly seen over a hundred miles away. Return to Manitou at 5:05 P. M. Those who do not care to make the Pike's Peak trip will find many interesting shorter trips from Manitou. Dinner will be served at the Cliff House, after which we will return via trolley to Colorado Springs, famous for its great wealth and ideal homes, where the evening will be spent. Sleeping cars will be ready for occupancy at 10:30 P. M.

SUNDAY, SEPTEMBER 19.—Leave Colorado Springs at 4:30 A. M. All meals during day in dining car. Everyone will be stirring early in the morning, as the train has purposely been scheduled to leave early, that most of the day may be spent in the regal Rockies. The famous Royal Gorge is passed at 7:45 A. M. This climax of mountain scenery is easily the grandest and most awe inspiring mountain defile ever mastered by railroad engineers. At its narrowest point the sheer half-mile-high walls of granite are about thirty feet apart where the train slips through on the much-talked-of hanging bridge, which carries the track over and parallel to the rushing stream around the projecting mountain wall. The memory of its stupendous proportions is everlasting. Beyond the route is one continuous moving picture of picturesque mountain scenery, staged on the most gigantic scale. The comeplling fascination of it all grows with the passage of each succeeding mile. The Continental Divide is crossed at Tennessee Pass, 10,240 feet high, through a half-mile tunnel, the western portal of which opens on the Pacific Slope. The eastern portion of Utah is passed during the night, and Salt Lake City is reached at 7:00 A. M. Surely there could be no more fitting way to spend the Sabbath than to thus pass through this wonderful panoramic scenery and view the Creator's handiwork.

MONDAY, SEPTEMBER 20.—Arrive at Salt Lake City at 7:00 A. M. Breakfast at Newhouse Hotel. This city, with its quaint Mormon institutions, its commercial aggressiveness and its attractive environs, has a fascinating lure to the tourist, and whose sojourn there is afterwards cherished among the most pleasant recollections of the entire trip. Here will be found the great Mormon Temple, the Tabernacle, the Bee Hives built by Brigham Young for his many wives, and other points of national interest. Leave Salt Lake City at 11:45 A. M., via the Western Pacific Railway. The route is through a country of beautiful and diversified scenery. Skirting the southern shores of Great Salt Lake, the salt beds of the desert basin are soon crossed. In places this salt is almost 12 feet thick and 98 per cent pure. As night draws nigh the sand and sage brush country is reached. Luncheon and dinner in dining car.

TUESDAY, SEPTEMBER 21.—This is the last, but by no means the least interesting, day of the trans-continental journey. As daylight approaches it brings the somewhat unusual experience of traveling through a region almost virgin in character, but with great possibilities which have heretofore lain dormant, because of remoteness from modern transportation facilities. Soon the foot-hills of the Sierras may be seen in the distance, and as the train proceeds the character of the country becomes more mountainous. At the summit Hawley is reached and the scenic attractions constantly increase. The Grand Canyon of the Feather River, a remarkable 90-mile-long gorge, through which the rails are laid on a shelf chiseled in the side of the canyon, and the alternating views of river, meadow, glen and valley, present a panorama which is indeed attractive. Oroville, at the western end of the canyon, is the center of a large gold-dredging district. Rolling down the western slope into the wealthy Sacramento Valley, the train passes through the city of the same name, which is California's capital and the center of a great fruit section. After passing through Stockton the train soon pulls into Oakland, from which point. San Francisco. our Convention City. is reached by ferry at

6:30 P. M. Breakfaast and luncheon will be served in the dining car, dinner in San Francisco.

RETURN ACCOMMODATIONS

In order to allow a freedom in the selection of return routes in accordance with the individual wishes of every member of the party, no definite special arrangements are made for the return trip, but return route *must* be decided upon at the time when your ticket is purchased. For convenience, however, au outline is given below of several of the most interesting and popular return routes. If it is found that the necessary number of persons will travel together in one party, special equipment will gladly be furnished, including any pre-arranged stop-overs that may be considered desirable.

ROUTE A.—Southern Pacific Railroad from San Francisco to Los Angeles, with side trip to San Diego, without extra transportation cost, but must be covered in ticket at time of purchase. Santa Fe Railroad from Los Angeles to Chicago, Denver or Kansas City. This will allow a side trip to the Grand Canyon of Arizona, from Williams, Ari., at an additional cost of $7.50.

ROUTE B.—Southern Pacific Railroad to Los Angeles, with side trip to San Diego, without extra transportation cost, and Southern Pacific Railroad from Los Angeles to New Orleans, etc.

ROUTE C.—The Great Northern Pacific Steamship Company to Portland, Ore., or Southern Pacific Railway to Portland. The Great Northern Railway to St. Paul via Tacoma, Seattle, Spokane, and Burlington Route from St. Paul to Chicago, etc., which takes you 300 miles along the picturesque Mississippi River. Side trip from San Francisco to Los Angeles may be included in ticket.

ROUTE D.—The Great Northern Pacific Steamship Company to Portland, Ore., or Southern Pacific Railway to Portland, Ore., and Northern Pacific Railway to St. Paul, Minnesota, via Tacoma, Seattle, Spokane, Burlington Route (Mississippi River scenic line) to Chicago. Side trip from San Francisco to Los Angeles may be included in ticket.

ROUTE E.—Southern Pacific Railway to Los Angeles, side trip to San Diego, without extra transportation cost. Salt Lake Route to Salt Lake City, Denver & Rio Grande Railroad. Colorado Midland Railway to Denver, Burlington Route to Chicago or St. Louis via Kansas City, St. Joseph or Omaha.

There are many other interesting and attractive return routes and your local ticket agent can give you full information regarding same.

BERTH RESERVATIONS AND GENERAL INFORMATION

As before stated, sleeping-car reservations should be made at an early date, in order that the necessary equipment may be reserved and comfortable accommodations insured for all. Reservations will be made in the order in which they are received. Further information about this special train, or as to fares, routes, reservations, etc., will be gladly and promptly furnished by W. H. Ross, Secretary, The American Society of Refrigerating Engineers, 154 Nassau Street, New York, N. Y., or by any Burlington Route representative.

A. S. R. E. Journal

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Published Bi-Monthly

by

The American Society of Refrigerating Engineers

154 Nassau Street, New York, N. Y.

Subscription price, United States and dependencies, $3.00 a year; Foreign countries in Postal Union, $4.00 a year.

Application for entry as second-class matter at the Post Office at New York, N. Y., pending.

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A. S. R. E. Journal

VOL. 2 SEPTEMBER, 1915 NO. 2

SAN BERNARDINO PRECOOLING PLANT

By C. M. GAY, LOS ANGELES, CAL.

Precooling, as related to transportation, is a term frequently misunderstood and misapplied. There is but little general experience in the art, and it has been confounded with cold storage and methods applied to static conditions from a misconception of its real application and significance. That it should appear as a complete, practical and efficient system of cooling applied to a whole train of loaded refrigerator cars, with a precursor of only a few sporadic experiments applied to single cars, is an evidence of careful and comprehensive engineering, supported by ample capital and earnest cooperation.

The term "precooling," properly applied, relates primarily to transportation and the immediate cooling of products within the refrigerator cars without rehandling and entirely apart from the idea of refrigerating in cold storage warehouses under ordinary practice. There is no element of novelty in the cooling of perishable products in refrigerated rooms, the ultimate object being either that of reduction of temperature only or the indefinite holding of products for an advanced market. This is a cooling or cold storage function entirely within the discretion and control of grower or shipper. Precooling, on the other hand, being performed within the refrigerator cars themselves, eliminates all question of cold storage or holding goods for market, and, by virtue of the fact that it is so performed, necessitates speed, eliminating every factor of delay as a prime requisite of successful operation and, therefore, logically becomes a function which is a prerogative of the transpor-

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tation companies, in the same manner that cold storage before delivery to the railway company is an unquestioned prerogative of the shipper.

The element of time involved in the operation of precooling being one of prime importance in accelerating the movement and availability of refrigerator cars and equipment furnishes a factor that fits exactly with the requirements of quick and efficient arrest of deterioration and decay in the fruit or other perishable products handled. The concrete advantages of precooling may be summarized as follows: Quick reduction of temperature to a safe point for transportation purposes; elimination of decay and deterioration; reduction of the number of stops and switches for re-icing in transit; shortening of time of delivery from the producer to market; avoidance of rehandling to and from storage, and extension of the market by the lengthened radius of safe shipment.

Granting the desirability, from the standpoint of the shipper, carrier and consumer, of securing these advantages, we are brought to the consideration of adequate precooling equipment, its cost and efficiency.

In 1907 the problem of the possibility of cooling the contents of a whole train of refrigerator cars loaded with oranges was assigned to the author by the Atchison, Topeka & Santa Fé Railroad Company. The idea was to accomplish in five hours or less a reduction in temperature of the fruit from the packing house temperature of 60° to 90° F. to the average refrigerator car temperature of 44° to 48°.

By agreement, an experimental refrigerating plant of the Santa Fé Railroad at San Bernardino, Cal., was turned over to the author for the purpose of ascertaining if it were possible to precool oranges in trainloads on track, and at what approximate cost and efficiency. The result of the experiments which followed and the engineering data compiled therefrom formed the basis of design upon which the present great Santa Fé plant was built at San Bernardino, Cal., a description of which will probably give a clearer idea of actual accomplishment, rather than a long treatise on what might be done.

The combined ice making and precooling plant of the Santa Fé Railroad is located in the center of the citrus fruit district at San Bernardino, and has been in continuous operation since its erection in 1910. "Pick-up" trains operated throughout the entire San Bernardino valley are taken to the precooling station as fast as loaded and, after a delay of about four hours for precooling, proceed to the eastern destinations, only occasional subsequent icing being needed en route.

The early experiments conducted by the author proved conclusively many engineering facts in regard to the condition of refrigerator cars and equipment, upon which were based inventions and devices peculiar to the system finally adopted. One fact stood out prominently—that all refrigerator cars were far from being airtight. Upon testing the cars, after closing them as tightly as possible, with an air pressure equivalent to half an inch, water gage,

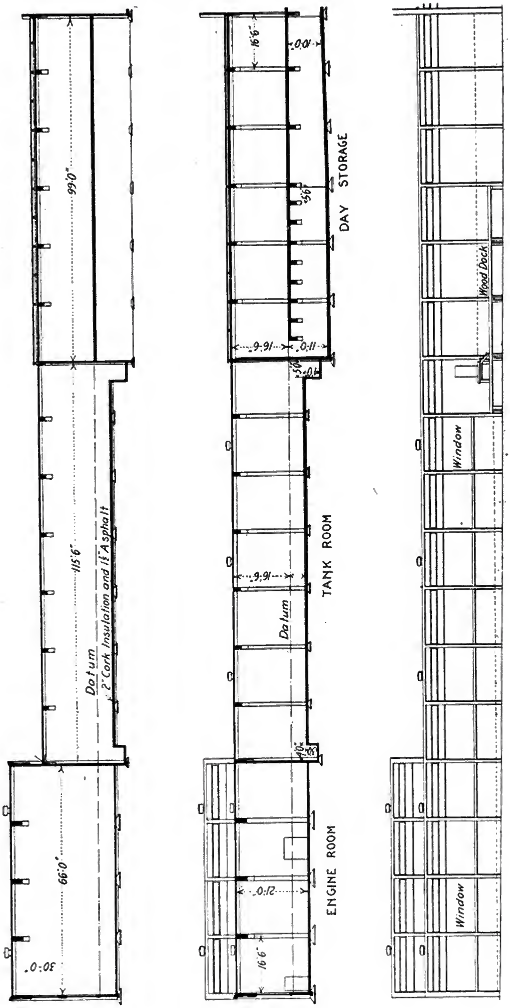

Fig. 1.—Longitudinal sections of boiler, engine, tank and day storage rooms.

they showed a loss of air from unseen openings of from eight hundred to two thousand cubic feet per minute, and an equivalent vacuum within the car produced practically the same result in the amount of air drawn in from the outside. Consequently, in order to perform the function of precooling in an efficient manner and without great loss of refrigeration, it was highly necessary that exactly as much air be withdrawn from the car as was being forced into it in the process of cooling, and that the air circulating system must be so balanced that the point of equalization between the pressure coming from the main pressure air ducts and the suction established in the main return ducts should occur as nearly as possible in the center of the car. This preliminary explanation is given in order to afford a better understanding of the description and illustrations.

The main building, Fig. 1, comprising the power plant, tankroom, day storage and winter storage, is 511 feet 6 inches long by 132 feet wide. Connected to the winter storage are the precooler chamber and icing dock, extending for a distance of 737 feet 6 inches, the icing dock proper being 693 feet long. This gives a full length of 1,249 feet for the plant, or approximately one-fourth of a mile. The structure is entirely of reinforced concrete, with interior partitions and tunnels of the same material. The building may be considered as divided into two portions, one devoted to ice manufacture and the other to precooling. A reinforced concrete reservoir has been built, partially underground, 30 feet away from the power house end that is 80 feet square by 10 feet deep and is used in connection with an ammonia condensing system of the two-coil or double-pipe type. Adjoining this a pumphouse has been erected in which are installed two water circulating pumps for supplying water for the ammonia condensers. One pump is steam driven and the other electrically driven. There is a transformer station on a line with the pump house, and a crude oil tank, 20 by 30 by 10 feet, has been constructed similar to the water reservoir and parallel to the boiler room.

Water for the entire plant, including ice manufacture, is obtained from an artesian well, which is connected to the reservoir by a 14-inch steel pipe line. The pipe line is on the surface of the ground at the well end and enters a measuring weir at the basin end approximately four inches below the edge of the reservoir, giving a gradient of but one per cent. The flow from this well furnishes a supply of 500 gallons a minute, the normal flow being increased to this extent by an air lift, for which purpose a four-inch air line is placed in the same trench with the water line. The air compressors for this system are located in the boiler room.

The ice manufacturing system comprises three 350-horsepower Sterling boilers, set two in a battery and one single. These furnish steam to two 24 by 42-inch cross-compound Corliss engines, each direct connected to one 300-ton, 17 by 34-inch duplex, double-acting Vilter refrigerating machine. Adjacent to the engine room is a 36 by 24-foot fore-cooler tank, also containing filters and ammonia receivers, the latter being placed under the filtering apparatus. The freezing tank chamber is 115 feet 6 inches long, and extends across the full width of the building, being 21 feet high in the ceiling, with the floor line below grade the distance noted. The floor is of concrete, covered with two-inch corkboard over which there is a finish of 1½ inches of asphalt. Three 75-ton icemaking tanks are here installed, giving a total capacity of 225 tons. The cans used are of the regulation size, 11 by 22 by 44 inches, forming 300-pound blocks, and are handled three at a time by electrically driven traveling cranes. One 300-ton refrigerating tank is used for the precooling system and is equipped with a Vilter flooded ammonia system of brine coils, with an accumulator.

The day storage room, Fig. 2, has a daily capacity of 900 tons of ice, when blocks of 300 pounds each are placed on end in one tier. The winter storage is divided into four compartments by six-inch concrete partitions, giving a storage capacity of about 7,000 tons for each section, Fig. 3. Each room in the winter storage is provided with an endless-chain ice elevator for storing, and has an iron ladder for access. Adjacent to the wall separating the tank room and the day storage, and below the floor of the latter, is a 49 by 33 by 11-foot brine chamber, which is shown in an accompanying drawing. This compartment contains a 200-ton, two-coil, double-pipe brine cooler and three electrically driven circulating pumps. The latter are interchangeable, two being used for the storage plant or two for the precooling system, as the conditions necessitate; by this arrangement it was possible to omit a fourth pumping unit.

A concrete tunnel eight feet wide leading from this brine cooling chamber extends through both storage rooms. At the end of the winter storage the tunnel becomes a double-deck structure, the added section being of like size and construction as the lower duct. This lower tunnel is used for the brine mains to the winter storage, and to the precooler coil chamber; the upper portion gives access to the winter storage and carries an endless-chain ice conveyor which leads from and through the day storage. The entire storage plant, including the brine chamber, is insulated with 1½-inch double Nonpareil corkboard, with an asphalt finish of 1½ inches thick on the floors. The roof is insulated with Linofelt and air spaces. Neither room has any outside openings, all ice being handled through the day chamber and the central tunnel in the winter storage room.

Connected by concrete tunnels, the precooler coil chamber adjoins the winter storage. It is 44 feet 6 inches long by 48 feet wide and is insulated with 3-inch corkboard. The installation consists of a brine coil of 2-inch galvanized pipe, supported by 4-inch channel uprights and small angle brackets riveted thereto. Eight double-inlet Sirocco fans, each about 10 feet diameter, are arranged in two banks of four each. A set of four fans, two upper and two lower on each side, is driven by an 85-horsepower motor. One bank is used for blowing, while the other is for exhausting, each function being capable of execution by either set without reversing

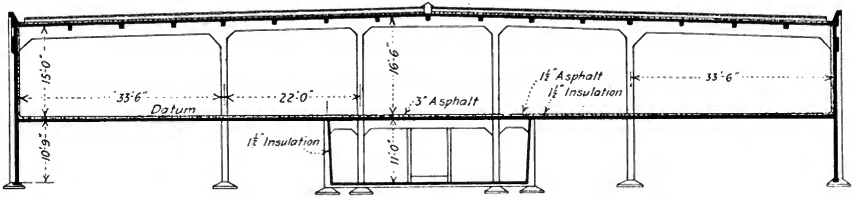

Fig. 2.—Transverse section through day storage room.

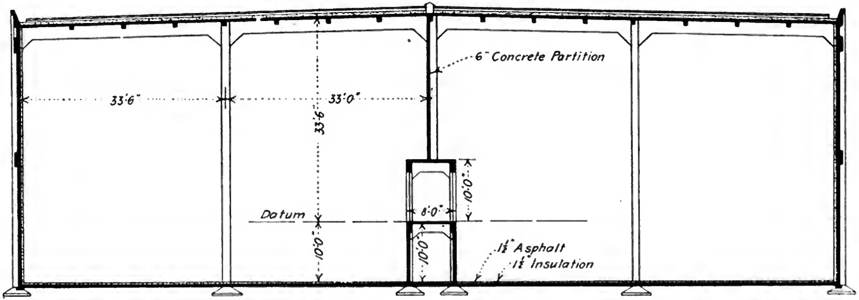

Fig. 3.—Transverse section through winter storage room.

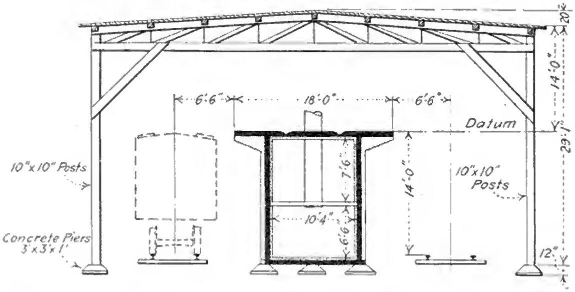

Fig. 4.—Transverse section through car shed, showing precooler air ducts.

the direction of rotation. A horizontal concrete baffle, literally a floor, is placed between the two banks.

Leading from the precooler chamber two longitudinal concrete tunnels or air ducts extend to form the main support of the car-icing dock. These are 7 feet 6 inches and 9 feet 6 inches high, respectively, and 10 feet 4 inches wide; both are of the same length, continuing the full length (93 feet) of the icing dock. The lower tunnel is the pressure duct, and the upper the vacuum duct, both being internally insulated with 3-inch corkboard. Refrigerating coils of 1¼-inch galvanzed iron brine pipe are arranged on both sides of either duct, a total length of about 50,000 feet being required. Connected to these sections are No. 18 galvanized iron pipes 22 inches internal diameter, with liquid sealed revolving joints. They serve as laterals to couple the system to the refrigerator cars, see Figs. 4, 5 and 6.

Purchasing electric current at high voltage from a central station, and stepping down to a voltage suitable for operation, the electric system offers no particular departure from ordinary practice; that is, the electrical energy is obtained at 10,000 volts and reduced to 440 volts for motor operation, and again to 110 volts for a three-wire lighting system. Alternation of steam driven and electrically driven auxiliary machinery permits of a continuance of operation should trouble occur at the central station.

The plant has a maximum precooling capacity of 32 cars, 16 on either side of the icing dock, in every four hours, or approximately 150 cars every 24 hours. The brine is taken from the 300-ton brine tank by the circulating pumps in the brine chamber beneath the day storage, sent through its refrigerating circuit and returned to be cooled by the brine coils in the same compartment. By this refrigeration of the main ducts the circulating air is kept in a similar state throughout its entire movement except when passing through the lateral connections to and from the cars. A large auto-

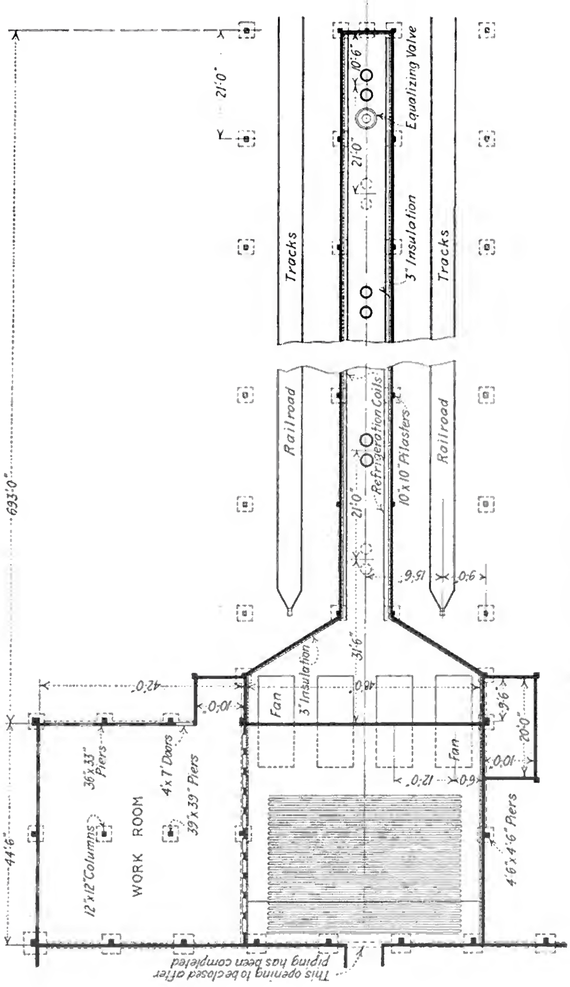

Fig. 5.—Sectional plan of air ducts and precooler.

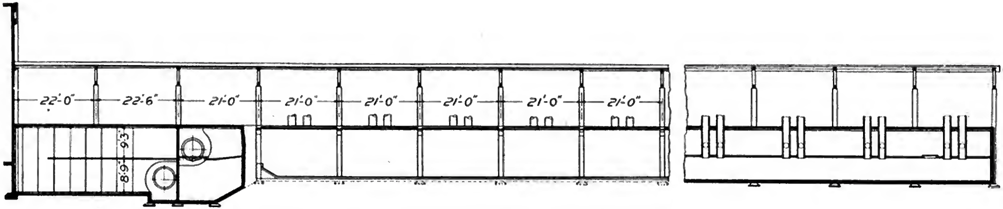

Fig. 6.—Longitudinal section, in elevation, through precooler air ducts.

matic equalizing valve connecting the pressure and vacuum ducts is placed near the terminals of the tunnels. By regulation of this valve the pressure of the air delivered to the cars and the suction at which it is withdrawn are positively governed. The opening of this valve forms a by-pass whenever the differential pressure becomes excessive, thus permitting the air to circulate through the main ducts and main cooling chamber even when no connections are made for precooling.

An automatic intake valve is located at the extreme outer end of the vacuum or low-pressure duct for the replacement of air which may be lost from the system by blowing the warm air from the cars. This is regulated to open when the vacuum falls below the differential pressure at which the equalizing valve is adjusted. The fresh air so taken into the system travels the full length of the suction duct, through the precooler coil chamber, and is then delivered to the high-pressure duct to be distributed through the cars. This insures the proper cooling of all air brought into the system from outside sources during the operation.

In the fan installation, the upper bank is used for exhaust from the low-pressure duct, delivering the air to the precooler coil chamber and thence over the brine coils. The other set draws the air from the coil chamber and discharges into the lower or main pressure duct for circulation through the system. The total air so handled is about 260,000 cubic feet per minute. The lower duct is held at a pressure corresponding to from one-half to three-quarters of an inch of water above atmosphere, and the upper at a vacuum of one-half to three-quarters of an inch of water. Accordingly, the air is supplied to the former at a temperature of from 8° to 10°, and is returned through the latter at a temperature varying from 20° to 24°.

The fundamental principle of the Gay system is to secure a balanced circuit with the pressure above atmosphere in the pressure duct, practically equivalent at all times to the vacuum below atmosphere in the return on the suction duct. By this method a balanced differential air pressure directly at the ducts is obtained without the overcoming of any frictional resistance in the coil chamber. The point of equalization of the air current is thus practically in the center of the refrigerator car, whenever such connections may be made. Tests at the plant show that the differential pressures are so equally balanced during precooling as to allow the opening of the refrigerator doors, in the center of each side of the car, without any apparent loss of air to the outside, or the bringing in of any warm air, although a current of from 6,000 to 8,000 cubic feet per minute of cold air is passing through the cars.

Valves fitted to each lateral pipe, opening into both the pressure and the vacuum ducts, as noted, secure the effect of cross connection and reversal of current, without any interchange of the laterals. By employing the liquid-sealed revolving joints, universal horizontal adjustment is obtained for connecting the laterals to the refrigerator car openings. In this manner the union can be made to an entire train without uncoupling or spotting of cars and regardless of their various lengths. The closing connections between the lateral air pipes and the cars is accomplished by a bellows spring and scissors-lever joint, comprised of leather and iron rings, operated by a crank. This preserves the entire area of the pipe under any adjustment and makes the connection airtight almost instantly. The pipes themselves are absolutely airtight, being externally insulated with two layers of three-quarter-inch felt suitably covered. By means of the universal joints they swing free of the car track when disconnected. No telescope connections are employed in this system and no internal valves, the valves used being visible for constant inspection.

Fig. 7 is a general view of the car icing dock, showing the ice conveyor in operation. Figs. 8 and 9, general views of the lateral air ducts, showing their means of suspension, and how they are swung into position and connected to the cars. Fig. 10, a sectional view showing details of construction of connections and valves through which the lateral ducts are placed in communication with the pressure and vacuum ducts. Fig. 11 shows a general view of the engine room.

In practice the cars are connected as shown. A funnel is placed over one bunker opening and the pressure air duct is first connected to the car for about twenty seconds. The cold air rushing in quickly displaces the warm air and vapors which are allowed to escape through the other bunker door at the opposite end. The lateral from the suction duct is then joined to the bunker at the other end of the car and circulation is established. During the cooler seasons of the year both connections are made simultaneously. The differential pressures, of course, control the speed with which the air passes; the violence of the current permeates to the center of every package, which is an essential consideration in proper precooling.

The total amount of refrigeration consumed by a car of fruit averages about three tons, depending upon the initial temperature of the fruit, which, varying with the season of the year, may cause this amount to become as low as one and one-half tons or as high as four tons of refrigeration. The amount of air circulating through each car in a period of four hours approximates one and one-half million cubic feet. When the plant is running at maximum capacity the amount of air circulated through each car per minute is about 8,000 cubic feet, with an average loss not exceeding five per cent. The cars are cooled to about 40° to 45° Fahrenheit. For icing the cars, upon the completion of precooling, or for such cars as are not precooled, the ice is handled by the endless-chain conveyor running through the day and winter storage. Here it may be diverted or carried on to the dock, the conveyor traversing the full length on both sides and making the ice accessible for immediate loading.

The plant is provided with facilities for metering all the brine used in precooling, and in cooling the ice-storage chambers, also with thermometers giving the temperatures at points of entering. Barometric columns are installed in the main air ducts and laterals,

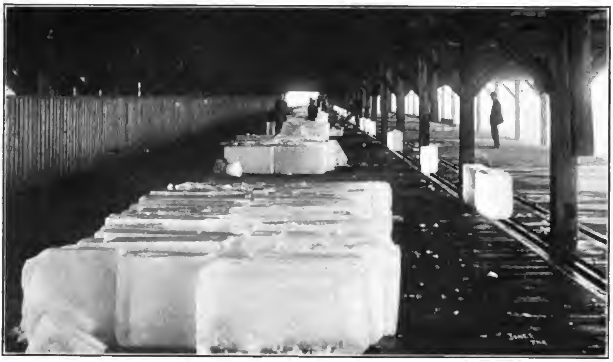

Fig. 7.—General view of car icing dock, showing ice conveyors.

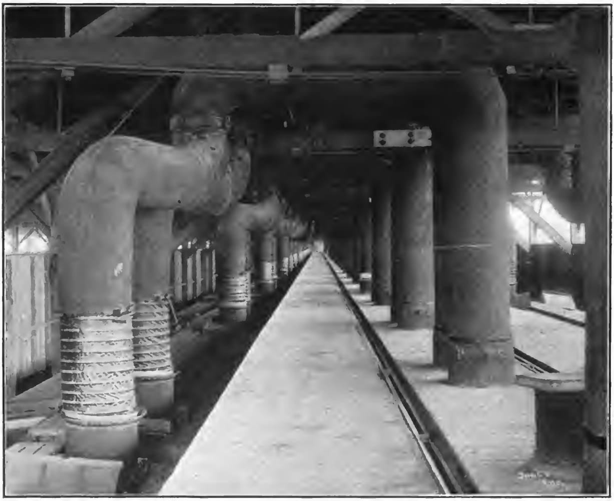

Fig. 9.—General view of car icing dock, showing lateral ducts.

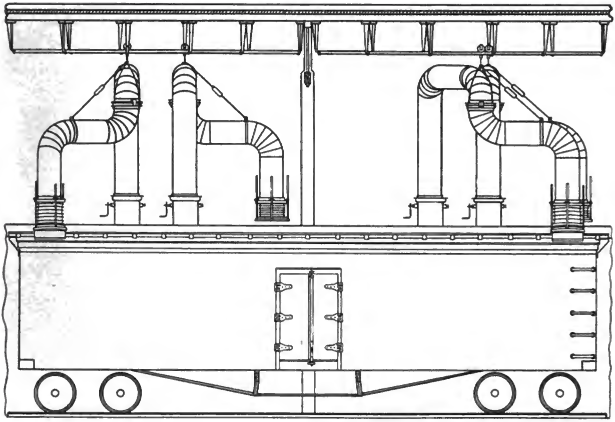

Fig. 8.—Elevation of car shed, showing means of attaching lateral ducts.

showing the pressure at which the air is circulated and returned. To afford a complete refrigerating record on every car handled, the velocity of air in both feed and return laterals is measured. A brine system of refrigeration has its advantages in precooling work. During the period of changing trains and other intermissions the plant is given an opportunity to store up cold brine, placing it in position to properly handle the large refrigerating demands made when a full load is incurred. The original charge of ammonia at this plant was about 13,000 pounds, and during its period of operation this has been added to but slightly.

The plant represents an investment of about $900,000. The special features of the plant are the combination of icemaking and precooling systems in the same structure, the avoidance of loss of refrigerated air either by leaking out of the car or by drawing outside air into the car, and the application of identical treatment to one or more cars at the same time, with the addition of effecting the precooling service without any alterations in the standard refrigerator car.

A feature of the operation of this system is the facility with which the cooling pipes are defrosted in the precooling system. The large quantity of humid air results in an accumulation of frost upon the cooling pipes, which would in time frost the pipes together, so as to impede air circulation and refrigeration. In this particular system the accumulation is comparatively slow, as compared with the amount of air circulated, owing to the fact that very little fresh air is ever drawn into the system in normal opera-

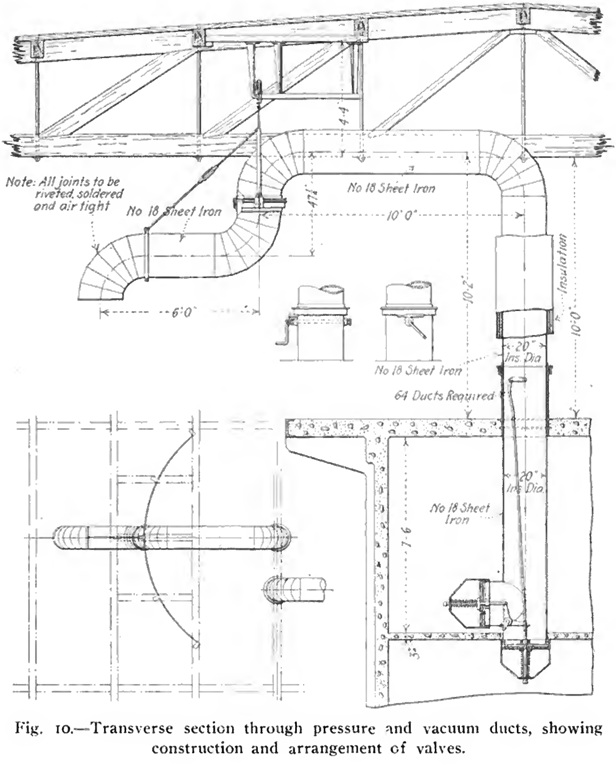

Fig. 10.—Transverse section through pressure and vacuum ducts, showing construction and arrangement of valves.

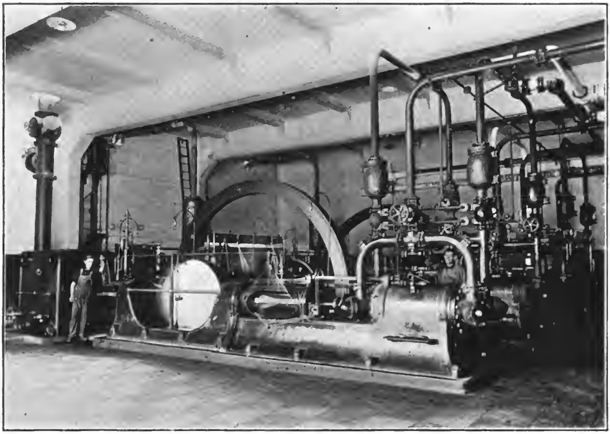

Fig. 11.—General view of engine room.

tion. However, some small degree of evaporation comes from the fruit itself in the short process of rapid air circulation in addition to the humidity of the contained air in the dfferent cars and in actual practice it is found advisable to defrost the coils at least once a week. This is accomplished by stopping the brine circulation and opening up four or five air laterals from the discharge or pressure duct to the outer atmosphere, when the great automatic equalizing valve will open under the suction from the fans, drawing the warm air from the outside, passing over the coils and out of the discharge laterals. With the brine circulation stopped and the warm external air being passed over the frosted coils and discharged again to the outside, it is found that one hour to one and one-half hours will completely remove the frost from the pipes and freezing surfaces. The sewer connections, which are charged with calcium in the sewer trap, are then opened and the melted frost discharged at once to the sewer, after which the connections are closed and the cooling pipes are ready for another week's operation without requiring defrosting.

Another feature of the Gay system is the arrangement for utilizing the precooling connections as preheating connections during severely cold weather. These are not installed or required at the San Bernardino plant, but may be installed in such precooling plants as are located where the winter temperatures are very severe, operating the cooling pipes with warm brine, producing a higher temperature of air to be circulated through the cars in exactly the same manner that the low temperature of air secures precooling.

The time consumed at this plant in connecting up two strings of fifteen cars each to the 22-inch air ducts and establishing complete circulation with four operatives employed in the work of making connections is only ten minutes, and the carrying temperatures are reached in the period of four hours or less, as compared with twenty-four to forty hours under ordinary cold storage conditions.

Since the establishment of this great plant over 28,000 cars have been precooled, with the result that damage claims have been practically eliminated on shipments precooled by the Santa Fé Railroad.

THE OIL ENGINE FOR ICE PLANT SERVICE*

By Louis K. Dolling, New York, N. Y.

During the past ten years internal combustion engines have been so perfected in design and construction that they now successfully compete with steam engines in all classes of work. Their reliability is such that ocean-going steamships are built with single units and submarines equipped with only high speed oil engines undertake cruises of 4,000 miles.

The internal combustion engine has been introduced on a larger scale in refrigerating plants only in the last three years. The reason is probably due to the fact that in the past nearly every plant that required refrigeration also needed steam. Packing houses and breweries required great quantities of heat and, until a few years ago, the majority of ice plants depended entirely on distilled water. The necessity of having a steam boiler, and the fact that by using exhaust steam power was available as a by-product, made the steam engine the proper prime mover. Another factor which retarded the introduction of the oil engine was the high initial cost. The price of high grade oil engines has been higher than that of steam plants with Corliss engines and low first cost is the main consideration for many refrigerating plants. Conditions have changed. Steam for heating purposes is now used more economically. The raw water ice plant has been perfected. Oil engines are being built better, cheaper and of higher economy. The cost of coal is rising, whereas oil is cheaper than it has ever been before.

A short comparison will show the superiority of the oil engine over the steam engine. An oil engine of an improved type develops under normal conditions one brake horsepower hour with 0.45 pounds of oil, which at 19,000 British thermal units per pound equals 8,500 British thermal units. This represents a brake horsepower thermal efficiency of 30 per cent. The best compound condensing, poppet valve, steam engine operated on a pressure of 150 pounds, with 100 degrees superheat and high vacuum, will produce an indicated horsepower hour on 12 pounds of steam. A boiler under such conditions will evaporate 8 pounds of water per pound of coal. The coal consumption per indicated horsepower hour is therefore 1.5 pounds and per brake horsepower hour about 1.66 pounds, which at 13,000 British thermal units per pound represents 21,500 British thermal units or 12 per cent thermal efficiency. The

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thermal efficiency of the oil engine is, therefore, 2½ times that of the steam engine.

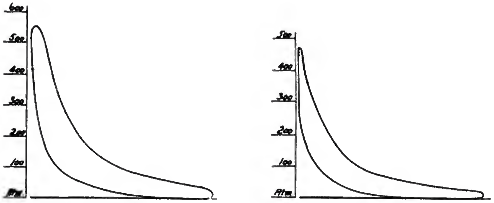
The desirability of installing a certain kind of engine, however, cannot be determined by thermal efficiency alone. Such other factors as relative cost of installation, cost of different fuels, etc., must also be considered. It is, nevertheless, interesting to see that the actual thermal efficiency of the oil engine is very close to the theoretical maximum efficiency of the steam plant.

Oil engines used in refrigerating plants are of the single-acting type and have capacities up to about 300 horsepower. They may be divided into four classes:

1. High compression or Diesel type engines. (Using air injection.)
2. Medium compression engines. (Using air injection.)
3. Four-cycle low-compression engines. (Using pump injection.)
4. Two-cycle crank-case scavenging and one-to-one scavenging engines. (Using pump injection with water.)

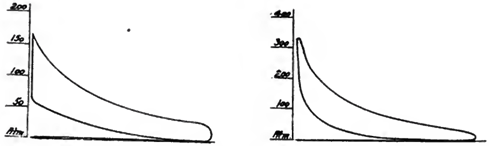
The working process of the Diesel engines is as follows: A charge of air is drawn into the cylinder on the suction stroke and is compressed in the working cylinder to from 500 to 600 pounds. The temperature of the air is thus raised above the ignition point of the fuel oil by compression. A few degrees before the end of the compression stroke oil is injected by means of air under from 1,000 to 1,200 pounds pressure per square inch, igniting as it enters the cylinder. The oil is admitted gradually, with the object of preventing a rise in pressure as the piston moves forward. On the completion of about 8 per cent of the stroke the spray valve closes, the combustion ceases, and the high temperature charge expands doing work on the piston until the end of the stroke is reached. At the end of the stroke the exhaust valve opens, the pressure drops and the piston returns, sweeping out the burned charge. It is the tendency to so control the fuel injection that the combustion will take place at constant pressure. The indicator card, Fig. 1, would then be flat on top to the point where the expansion curve begins, and for this reason the Diesel is sometimes called an equal pressure engine. This expression, however, is not quite accurate, for in practice there is always a rise in pressure of from 100 to 150 pounds on account of the difficulty of regulating the injection perfectly and obtaining good combustion. Diesel engines are nearly all of the four-cycle type, which experience has proved to be the better. The two-cycle type involves greater difficulties, due to the higher temperatures to which the cylinders and heads are subjected, on account of there being double the number of ignitions per minute. At the same time, the two-cycle principle does not simplify the design nor decrease the cost materially. Scavenging air in excess of the piston displacement is necessary, demanding an expensive separate scavenging pump, and even under the best conditions the mean effective pressure is greatly reduced.

Diesel engines are built both vertically and horizontally. The



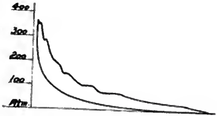
Diesel type.

Medium-compression type.



Four-cycle, iow-compression type (HA).

Four-cycle, low-compression type (DH).



Two-cycle, pump-injection type.

Fig. 1.—Indicator cards for different types of oil engines.

vertical type makes the design of multiple-cylinder engines easier and requires less floor space; while the attendance factor becomes simpler with the horizontal construction. Both types are equally reliable and economical, the selection of one or the other being a matter of taste and of local conditions. The starting of a Diesel engine is generally effected by compressed air of at least 1,000 pounds pressure per square inch, the high pressure being needed to inject the fuel and to compress the first charge of air to the pressure needed for ignition. If, however, a sufficient quantity of air is available to bring the engine up to a high speed, a pressure of 200 pounds per square inch may be sufficient. In such case the compression is relieved until the speed is sufficient to carry the crank over against full compression. The mean effective pressure on a brake horsepower basis is about 70 pounds per square inch.

Fig. 2 shows a vertical Diesel engine as built by the Busch-Sulzer Bros.-Diesel Engine Company in St. Louis, the oldest manufacturers of Diesel engines in the United States and the original owners of the Diesel patents. Other manufacturers of vertical Diesel engines in this country are the Fulton Iron Works, St. Louis, Mo.; McIntosh & Seymour, Auburn, N. Y.; Lyons Atlas Company, Indianapolis, Ind.; the Dow Pump & Diesel Engine Company, San Francisco, Cal., and the New London Ship & Engine Company, Groton, Conn., which makes a specialty of engines for marine purposes. The General Electric Company has also built several high speed Diesel engines for direct connection to generators. Horizontal Diesel engines as shown in Fig 3 are built by the Allis-Chalmers Company, Milwaukee, Wis., and the Snow Steam Pump Company, Buffalo, N. Y. The Nordberg Manufacturing Company also builds a two-cycle one-to-one scavenging high-compression engine, which, however, does not employ air for the injection of the fuel.

The medium-pressure oil engine compresses the air charge to from 250 to 300 pounds per square inch, which is not sufficient to ignite the infected fuel. Ignition is effected by a hot surface in the cylinder head against which the fuel is sprayed, the pressure rising to from 400 to 500 pounds per square inch. These engines are of the constant volume type, in which combustion takes place at approximately constant volume. Indicator cards from these engines are therefore similar to those of gasoline or gas engines and, in fact, almost identical in form to steam-engine cards with very early cut-offs. The mean effective pressure is 64 pounds per square inch on the basis of brake horsepower. Constant volume combustion is the most economical, and for that reason this type of engine is as efficient as the Diesel engine, although the compression is only about half as severe. Before this type of engine can be started, the ignition plate must be heated by an oil or gas torch, which will require from 1½ to 15 minutes, according to the efficiency of the torch. As the engine does not depend on the heat of compression for ignition, the compression is reduced during starting, so that air pressure of from 100 to 150 pounds is quite sufficient. Fig. 4 is a horizontal engine of the medium-compression type, built by the

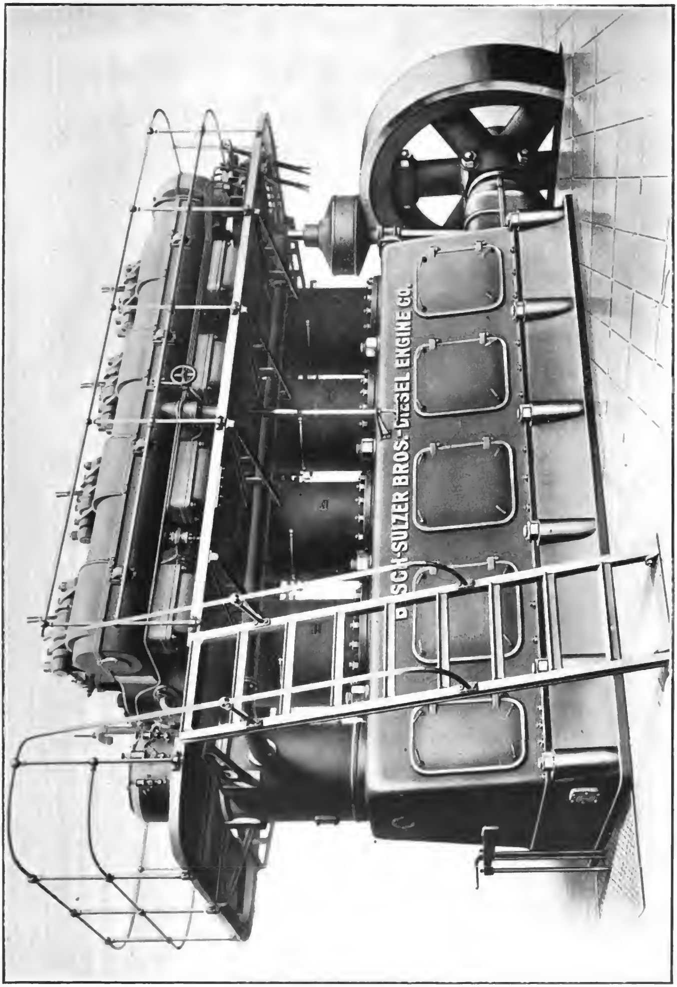


Fig. 2.—Busch-Sulzer Brothers-Diesel Engine.

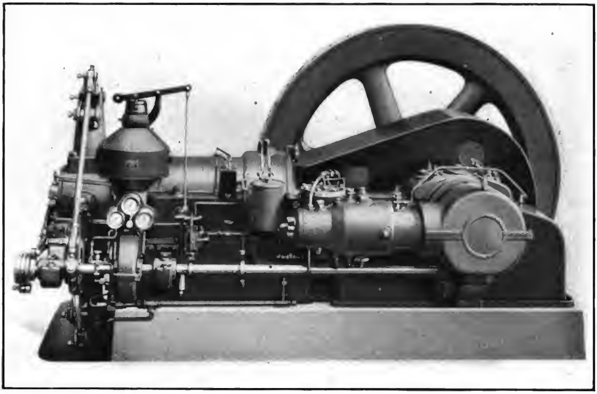


Fig. 3.—Allis-Chalmers Horizontal Diesel Engine.

De La Vergne Machine Company of New York and known as the type "FH."

Diesel and medium-compression engines burn all commercial oils obtainable in the United States. Very heavy oils, however, with a high point of ignition, cause more difficulty in Diesel engines than in medium-compression engines, because the hot vaporizes of the latter insures the ignition under all circumstances. The oil consumption per brake horsepower is about 0.45 pounds at full load, 0.46 at three-quarters load, 0.52 at one-half load, 0.7 at one-quarter load and about 9 pounds per hour for a 100-horsepower engine at no load. At 10 per cent overload the consumption is approximately 0.5 pounds per brake horsepower hour. Fig. 5 shows the oil consumption at different loads. It will be noted that the curve is very flat, showing the small change in economy at the various horsepowers. This illustrates one great advantage of the oil engine over other prime movers.

Four-cycle, low-compression oil engines work with compressions of from 60 to 180 pounds per square inch. They inject and spray the fuel mechanically into the combustion chamber, where it is ignited by means of a hot surface. An early type of four-cycle, low-compression oil engine is the Hornsby Akroyd or De La Vergne type "HA," as shown in Fig. 6. The compression in this engine is only 60 pounds and the maximum pressure 180 pounds per square inch. The mean effective pressure is also low—about 90 pounds per square inch on a brake horsepower basis. The oil consumption is about 0.9 pounds per brake horsepower hour at from three-quarters to full load. This engine will not burn every commercial oil available.

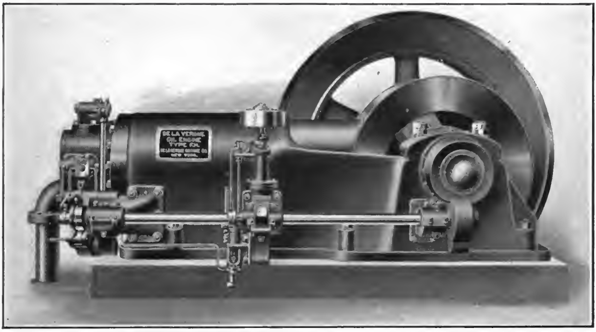


Fig. 4.—De La Vergne type "FH" medium-compression engine.

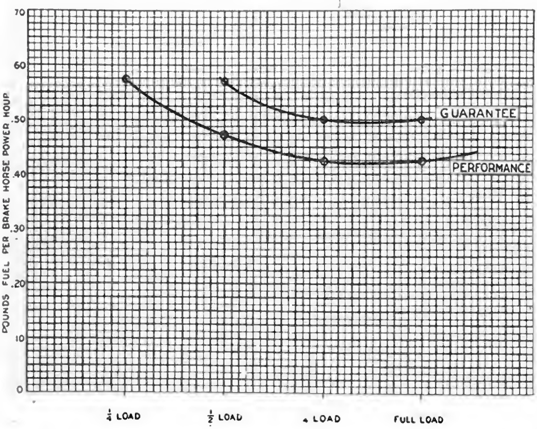


Fig. 5.—Fuel consumption per B. H. P. hour for Diesel and medium-compression engines.

but requires a fuel of which a large per cent will distill off at comparatively low temperatures. Usually all oils above 28° Beaume are suitable for the "HA" engine, although some heavier oils burn with entire satisfaction, and some lighter oils are entirely unsuitable. With this type of engine it is necessary to remove the vaporizer cap about once a week to remove the carbon deposit. A more highly developed four-cycle, low-compression engine is the De La Vergne type "DH," shown in Fig. 7. With this engine, the compression pressure is about 150 pounds per square inch and the maximum pressure from 280 to 300 pounds per square inch. The mean effective pressure is approximately 57 pounds per square inch on a brake horsepower basis. The consumption at full load is about one-half pound per brake horsepower hour, and at three-quarters load it is about the same. So far, this four-cycle, pump-injection engine has worked satisfactorily on every grade of oil that it has been offered for use.

Two-cycle, pump-injection engines are built in two general types: First, the crank-case scavenging type, of which the Mietz & Weiss engine, Fig. 8, is an example. Quite a number of different engines of this type are manufactured in the United States. The Muncie Oil Engine, the Remington, the Venn Severn, the Crescent and many others are built on this crank-case scavenging design. Another type of two-cycle, pump-injection engine is the one-to-one scavenging type, of which the Bessemer engine, shown in Fig. 9 is a good example.

Two-cycle, pump-injection engines are mostly light built, high speed units and are therefore seldom used in connection with large refrigerating installations. Some of this type of engine will operate on oils as heavy as 24° Beaume, while many of them require lighter distillates. All of the two-cycle, pump-injection engines employ water injection to control cylinder temperatures, this being distinctly characteristic of the two-cycle, pump-injection equipment.

A comparison of the different types of engines gives the following characteristics: The high compression or Diesel engine is highly economical and well developed in design and construction. On account of the high pressures and temperatures, it must be built of special material and with first-class workmanship. Consequently, the first cost is high, and to avoid rapid wear and tear the attendance must be of a high order. As ignition depends entirely on compression, a blowing piston or a leaky valve can stop the engine. If sufficient compressed air is available, the engine can be started immediately, which is an advantage in some cases, such as where the engine is used for fire service, for instance.

The medium-compression engine shows the same economy as the Diesel engine, and the first cost is about the same. The attendance should also be high class, but as the ignition is caused by a hot surface and not by compression, and as the compression and average pressures are considerably lower, the engine is not so sensitive to abuse. That the vaporizer must be heated before starting is a disadvantage compared with the Diesel; but in most cases it is of no

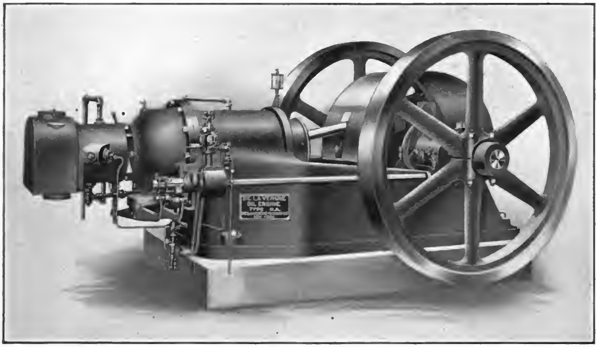


Fig. 6.—De La Vergne type "HA" oil engine.

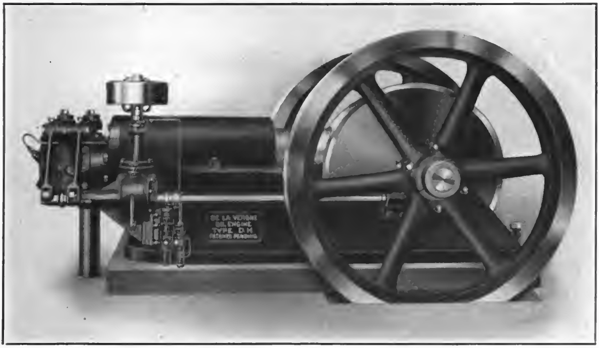


Fig. 7.—De La Vergne type "DH" oil engine

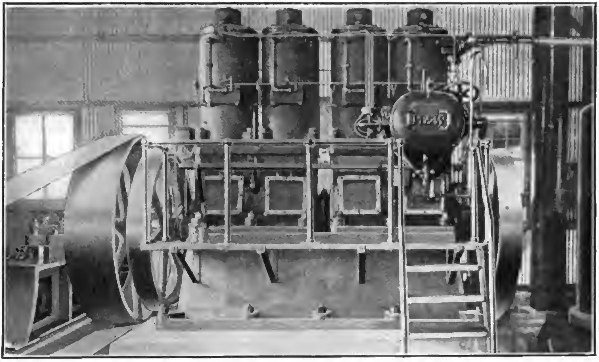


Fig. 8.—Mietz & Weiss two-cycle, crank-case scavenging engine

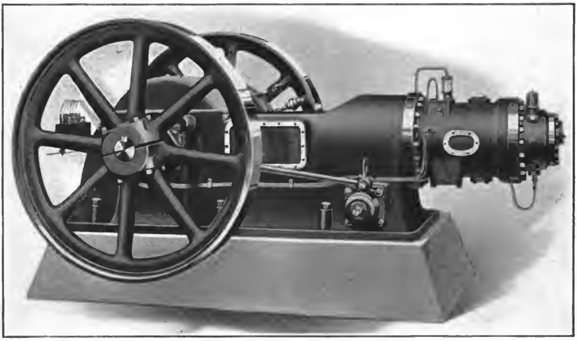


Fig. 9.—Bessemer horizontal, two-cycle, one-to-one scavenging engine.

consequence, as the heating can be accomplished while the engineer prepares the engine for starting. The latest designed torches make a two-minute start entirely practicable.

The latest type of four-cycle, low-compression engine consumes 15 per cent more fuel than the Diesel and the medium-compression engines, but they are simpler in design and construction and considerably cheaper in first cost. No high-pressure air compressor for the fuel injection is needed, and the low pressures and temperatures make attendance a much simpler matter. In many cases engines of this type are preferable, especially in sizes up to 150 or 200 horsepower where fuel oil is not expensive, or where the engines run only a part of the year.

Four-cycle, low-compression engines and two-cycle, pump-injection engines up to 50 horsepower can be started by hand, whereas larger units require air at about 75 pounds per square inch pressure. Medium-compression units require a starting air pressure of about 100 pounds per square inch. Diesel engines are usually started on air of 1,000 pounds per square inch, although by the use of large storage tanks it is possible to start with a pressure of about 200 pounds per square inch. The air is stored in one or two tanks of large enough capacity to start the equipment at least three times, the exact volume of air depending on the type of engine and the pressure at which it is stored. Oil engines are arranged to maintain their own supply of starting air, although some means must be provided to obtain the charge independently. All oil-engine-driven plants of above 50 horsepower capacity should be equipped with a dependable air compressor driven by a gas or gasoline engine or electric motor for obtaining the starting air initially, or in case of emergency. It is a great mistake to install a compressor of too small capacity, as this results in a serious loss of time whenever it is necessary to have a charge of starting air.

When installing an oil engine to drive a refrigerating machine, it should be remembered that the oil engine differs from the steam engine in several respects. The first and most important difference is the impossibility of overloading the oil engine appreciably for long periods. The oil engine is the equivalent of the whole steam plant, boiler and engine combined. Overloading oil engines produces results similar to forcing boilers, but in an aggravated form, because the temperature and pressures in the oil engine cylinder are higher. A boiler that is forced for a considerable time shows excessive wear and finally fails. In an oil engine running at overload, the head, piston and valves suffer, entailing expensive repairs. It is therefore essential to select an engine of such size that no overloading will occur under the highest condenser pressure and back pressure. Ninety per cent of all difficulties with oil engines in ice-plant service are due to overloading. It should not be understood that an oil engine must never be overloaded. On the contrary, most oil engines will carry from 20 to 25 per cent overload, and such an overload for an hour or two will do no damage. In an ice plant, however, if there is any overload at all, it will usually last for days.

Another great difference between the oil engine and steam engine is in the rotative speed at rated capacity. Oil engines usually run at such a speed that the compressor cannot be directly coupled, but must be connected through a belt or rope drive. The installation of such drives needs careful consideration and the losses of transmission are greater than generally supposed.

As the engines are single acting, and mostly of the four-cycle type, their operation cannot be expected to be quite as smooth and regular as that of steam engines, and, since both the prime mover and the compressor have uneven crank efforts and both have heavy flywheels, the transmission should be at least twice as heavy as though a steam engine were used, and four times heavier than if the torque were uniform. To pulleys, clutches and couplings the same rules apply; they should be selected with two to four times the capacity listed by the manufacturers. The first cost will be higher, but much trouble and annoyance will be saved. With a rope drive the grooves should be deep, to avoid the possibility of the ropes jumping out—a possibility none too remote. Fig. 10 and Fig. 11 illustrate two oil engine-driven ice plants.

On account of the high temperatures during combustion, cylinder heads and liners must be effectively water cooled. It is, therefore, necessary to provide for an uninterrupted supply of a sufficient quantity of pure, clean water. Failure of the water supply or the use of dirty or hard water has an effect similar to that in a steam plant. The oil engine does not explode, like a boiler, when it runs without water or when heavy scale is formed, but the cylinder head is liable to crack, valves will leak and cylinder liners will wear out quickly. A great

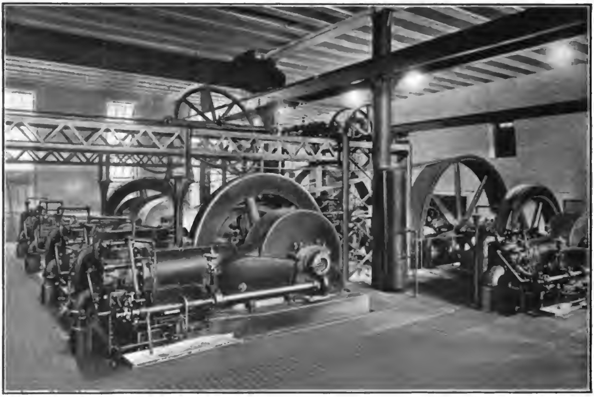


Fig. 10.—200-ton oil engine driven ice plant.

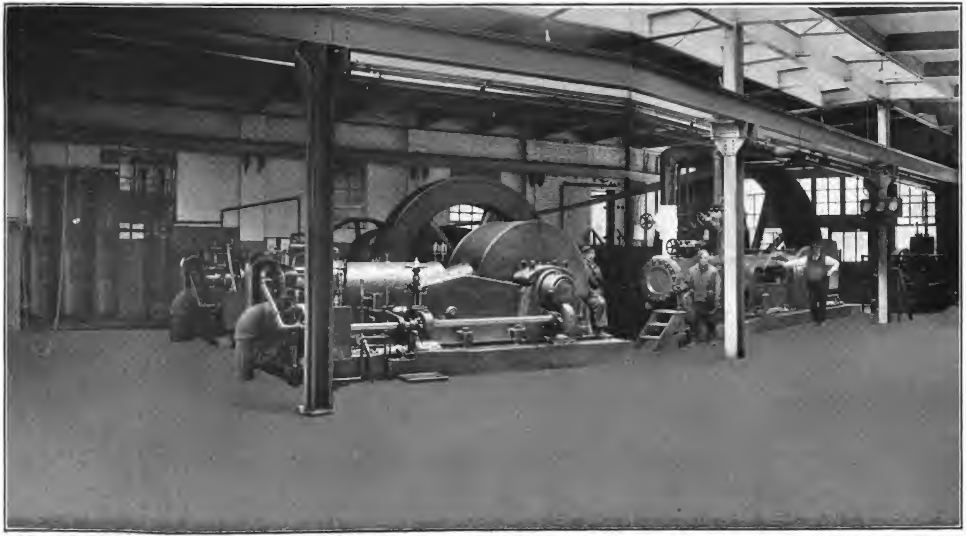


Fig. 11.—Oil engine driven ice plant, Union Ice Company, Oakland, Cal.

number of the cracked cylinder heads and worn-out pistons that are charged against the manufacturer are caused by faulty water supply. Here is another difference between steam plants and oil engines. When a boiler explodes there is seldom any qustion about placing the responsibility to the neglect of the engineer, but with oil engines it is the reverse. Every breakdown is placed to the debit of the engine, and the manufacturer is invited to prove that the trouble was due to carelessness—a very difficult matter in many cases.

With an oil-engine plant the cooling water for the engine is usually taken from the condenser discharge. But if there is much hardness in the water a separate cooling system should be installed; and as the quantity of water is not great, and as it is sufficient to cool the water to from about 90° to 100° F., the cooling plant is small and inexpensive. The amount of heat to be carried off by the cooling water is about 2,500 British thermal units per brake horsepower hour. When the cooling water enters the engine at 65° F. and leaves it at 180° F. 3 gallons of water will be required per brake horsepower hour for Diesel and medium compression engines. When the water is taken from the condensers at a temperature of about 90°, five gallons per brake horsepower hour should be provided for, while, with a cooling plant, from about 8 to 10 gallons of water should be circulated per brake horsepower hour. The evaporation with such a cooling system is from ¼ to ½ gallon per brake horsepower hour, so the size of the cooling plant can easily be determined. Fig. 12 shows a simple and inexpensive, yet effective and suitable recooling plant for oil engines. If conditions allow it, rainwater should be collected in the tank, as it is free from scale-forming salts. It should be borne in mind that all the hardness introduced with the make-up water is ultimately deposited in the cylinder head usually at the places where the most effective cooling is needed. In arid regions where soft water is not available a closed system is sometimes employed.

A good arrangement of the exhaust piping is essential. The exhaust valve opens when the pressure in the cylinder is from 30 to 45 pounds; the temperature of the gases immediately after leaving the cylinder is from 700° to 900° F. and the pressure in a properly arranged exhaust pipe should not be over about 3 pounds. As the discharge is intermittent and the velocity of the out-flowing gases very high, the exhaust is noisy and it is necessary to carefully provide means for carrying off the gases and for muffling the noise. In placing the exhaust piping, provision must be made for considerable expansion due to the high temperature, and the piping must be kept away from inflammable material. It is advisable to have that part of the exhaust piping near to the engine water jacketed and the piping placed in trenches, to avoid overheating the room. The piping may be of steel, but in cases where the oil contains much sulphur and the exhaust piping is so long that the steam contained in the exhaust has a chance to condense, cast iron pipe should be used. The sulphur dioxide in the exhaust combines with the water of combustion to form sulphuric acid, which destroys steel piping

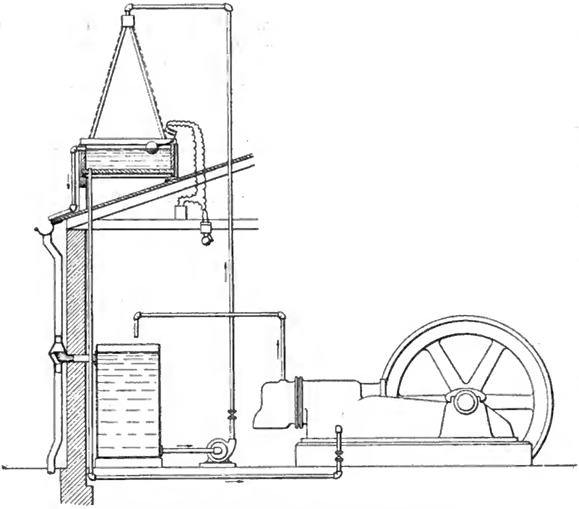


Fig. 12.—Precooling system for an oil plant.

sometimes in a few months. In most cases, however, steel piping is sufficient and will last 10 to 15 years.

The end of the exhaust pipe should be located so as to be visible from the engine room, for the condition of the exhaust is an index of condition of the engine. The exhaust of Diesel, medium compression and type "DH" engines running in good condition should be clear. A smoky, dark exhaust gives warning that something is wrong; either the engine is overloaded, the valves are not tight or the fuel injection is defective. In such case the engine should be shut down as soon as possible and the defect corrected, for if this is not done more serious trouble will develop sooner or later, possibly months afterward, but the trouble will come eventually. If the locality makes it impossible to place the end of the exhaust pipe visible to the engineer, a test cock should be placed in the exhaust pipe, near the engine at some convenient place, so that the engineer may open it periodically and observe the color of the exhaust. This, however, is not as satisfactory as viewing the full discharge from the end of the exhaust pipe.

The noise of the exhaust can be muffled by placing a tank of three to four times the cylinder volume near the engine. The tank should be made of cast iron, because the vibration of a steel tank produces a ringing, metallic sound unless it is coated with asbestos cement or other covering. In localities where the exhaust must be noiseless a second muffler may be installed some distance from the first one. It is necessary to provide each engine with a separate exhaust and, preferably, to provide a separate pipe for each cylinder. Occasionally several units of a plant have been connected into one big exhaust stack; but the results are not satisfactory, because inlet and exhaust valves are generally open simultaneously for from about 40 to 50°, so that the charge of one engine may be so influenced by the exhaust of another as to result in a reduction of power. With such an arrangement it is impossible to determine which engine gives the smoky exhaust.

About 2,500 British thermal units per brake horsepower hour are lost in the exhaust gases. It is tempting to make use of this heat by passing the gas through a steam boiler, but, as the temperature of the gases is comparatively low, only about one-third of the waste heat can be utilized unless the boiler be made very large. By using discharged cooling water to feed the exhaust boiler about 1 pound of steam can be developed per brake horsepower hour. In ice plants, however, the cooling water is usually valuable for thawing off the cans or plates. As the production of steam is dependent upon the running of the engine and upon its load, an exhaust boiler is seldom desirable. The complication and cost connected with such an installation in most cases outweighs the saving. It is different, however, where hot water can be used instead of steam. By inserting a water heater in the exhaust line and using the cooling water from 3 to 4 gallons per brake horsepower hour of water of 200° can be made available. But, as the cooling water of the engine alone is sufficient for the thawing off of the ice cans, such a boiler is advisable only if hot water is needed for other purposes.

One of the great advantages of plants driven by oil engines is the easy, automatic manner in which the fuel is handled and the absence of ashes and clinkers and all the troubles connected with the disposing of same. In arranging the fuel storage and supply it is advisable to observe certain rules found by experience or established by the National Board of Fire Underwriters. The next, Fig. 13, shows a typical fuel oil system. The storage tanks should be made of steel and should be large enough to hold one month's fuel supply. If the oil can be delivered in tank cars the storage tanks should have a capacity of from 10,000 to 12,000 gallons each, and it is desirable to have two tanks. They should preferably be placed underground for protection from fire, to facilitate filling and to allow the oil in the engine room to drain back to the tanks.

Oil engine lubrication should have the same attention as that of a high speed steam engine. As in the case of steam engines, oil engine cylinders require a different oil from the bearings, a high flash point being desirable on account of the high temperatures, and a low viscosity being desirable in that the piston may be washed free of carbon particles. The cylinder oil should be delivered by a force feed. Too much cylinder oil not only means an unnecessary waste,

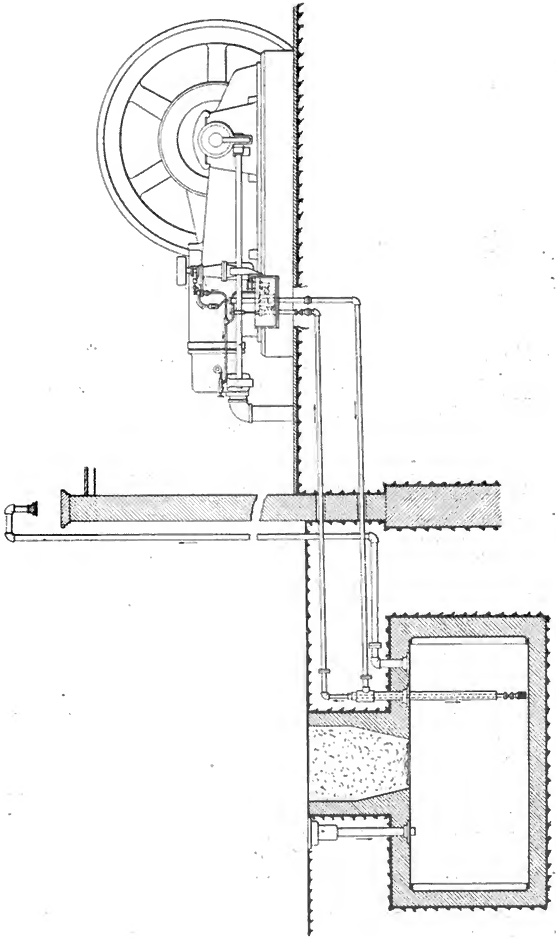


Fig. 13.—Fuel oil storage system.

but is actually harmful, as the superfluous oil burns, forming carbon, resulting in excessive friction and cylinder wear. Too little oil, on the other hand, means heavy wear or cutting of the cylinder and piston.

A most suitable oil for cylinder lubrication is liquid base oil, which has the following characteristics: Gravity, 25.8° Beaume; specific gravity, 0.899; flash point = 345° F.; viscosity (Saybolt), 110 at 100°; cold test, —8° F. Liquid base oil is comparatively cheap, costing about 25 cents per gallon in New York. Any good engine oil may be used for the bearings, "Atlantic Red" being a very satisfactory lubricant for this purpose. It has the following characteristics: Gravity, 23.8° Beaume; specific gravity, 0.911; viscosity (Saybolt), 240 at 100°; flash point, 385-400° F.; cold test, 37° F. A suitable engine oil can be obtained for about 18 cents per gallon in New York.

An oil engine consumes about 0.75 gallons of lubricating oil per 1,000 rated horsepower hours, of which about one-third is cylinder oil and two-thirds machine oil. That makes the cost for the oil engine of a 100-ton ice plant at New York about $1.20 per day of 24 hours.

By filtering the oil, about 75 per cent of the machine oil used can be recovered and used for lubrication of the bearings, reducing the cost to about 66 cents per 100 tons of ice.

Fig. 14 shows the lubrication system adopted by the De La Vergne Machine Company, and Fig. 15, the arrangement of the oil reclaimer used in connection with this system. The object of the oil reclaimer is to remove the very fine particles of carbon from the oil, which experience shows cannot be separated by filtration, gravity settling or centrifugal action.

It is impossible to give strict rules for the installation of oil engines for ice-making plants, because the conditions and requirements of all plants are not alike, but for the most plants the following recommendations will apply:

1. Calculate the horsepower required for the compressor on the following assumptions: Back pressure, 20 pounds; head pressure, 220 pounds; exponent of compression, 1.3; mechanical efficiency of compressor, 85 per cent; efficiency of belt or rope drive, 85 per cent. With a can plant using raw water with air agitation having 15 cans per ton this will represent about 3¼ horsepower per ton of ice. For the auxiliaries the amount of power required will vary considerably, depending on the system used and the means of transmission, but care should be taken that the losses in the generator, in the electrical transmission and in the motors are liberally provided for. From 1 to 1¼ horsepower should be sufficient for the electrically driven auxiliaries of a can plant.

2. Drive compressor and auxiliaries from separate engines; for by so doing it is possible to change the speed of the compressor or to shut it down entirely. If the auxiliaries and compres-

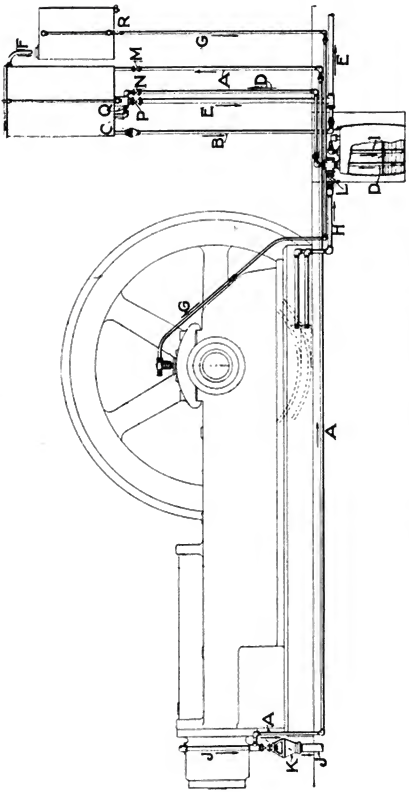


Fig. 14.—De La Vergne lubricating system.

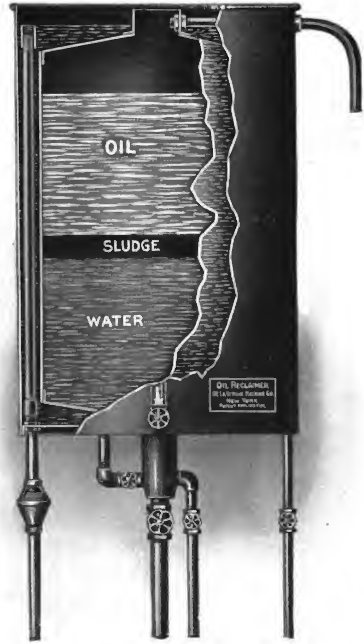


Fig. 15.—De La Vergne oil reclaimer.

sor are driven by the same engine, which is sometimes resorted to to make the cost of installation lower, the engine must always run to suit the generator speed or the speed of the auxiliaries.

3. Install more than one unit to make the plant flexible and provide against a total shutdown in case of accident. The approximate costs of steam- and oil-engine-driven ice plants of 50, 100 and 200 tons ice-making capacity are given in Table 1. The first cost of the oil-engine-driven ice plant is about 10 per cent higher than for a steam-driven plant. Table 2 shows the cost of producing ice by steam plants compared with oil-engine-driven plants. This shows that the cost is about 20 per cent greater for steam plants than for oil engine equipment.

Fig. 16 shows a rope-driven, 100-ton ice-making unit compared

TABLE 1.

COMPARISON OF INVESTMENT BETWEEN OIL ENGINE AND STEAM-DRIVEN ICE-MAKING PLANTS.

	Oil Engine Drive.			Steam Plants.		
Capacity every 24 hours	50 tons	100 tons	200 tons	50 tons	100 tons	200 tons
Investment:						
Oil engines, including rope drive and foundation	$14,500	$25,500	$45,000	......	......	......
Exhaust and water piping, oil storage tank, etc	1,500	2,500	4,500	......	......	......
Boiler plant and masonry	......	......	......	3,000	5,500	10,000
Steam, exhaust and water piping	......	......	......	1,500	2,000	3,000
Steam engine for direct connection, including foundation	......	......	......	5,000	9,500	15,500
Compressors, condensers and ammonia piping	5,500	10,000	18,000	5,500	10,000	18,000
Total cost compression side and power plant	$21,500	$38,000	$67,500	$15,000	$27,000	$46,500
Freezing system, complete	$14,000	$27,000	$50,000	$14,000	$27,000	$50,000
Ice storage	1,000	1,500	2,500	1,000	1,500	2,500
Building	16,000	35,000	60,000	16,000	35,000	60,000
Property	5,000	8,000	12,000	5,000	8,000	12,000
Total	$57,500	$109,500	$192,000	$51,000	$98,500	$171,000

TABLE 2.

COMPARISON OF DAILY AND YEARLY COST OF OPERATION BETWEEN OIL ENGINE AND STEAM DRIVEN ICE-MAKING PLANTS.

(ANNUAL LOAD FACTOR 60%)

	Oil Engine Drive.			Steam Plants.		
Capacity Every 24 Hours.	50 tons.	100 tons.	200 tons.	50 tons.	100 tons.	200 tons.
Daily Operating Expense:						
Chief ngineeer	$5.00	$6.00	$7.00	$4.00	$5.00	$6.00
Assistant engineers	3.00	3.50	4.00	3.00	3.50	4.00
Oilers	.....	4.00	4.00	.....	4.00	4.00
Firemen	.....	.....	.....	4.00	4.50	4.50
Tankmen	4.00	8.00	16.00	4.00	8.00	16.00
Storehousemen	4.00	4.00	4.00	4.00	4.00	4.00
Other labor	.....	.....	2.00	.....	.....	2.00
Fuel:						
Coal at $3.50 per ton	.....	.....	.....	22.75	38.50	77.00
Oil at 3½c. per gallon	7.88	15.00	30.00	.....	.....	.....
Ammonia, oil, waste, etc.	6.00	10.00	18.00	6.00	10.00	18.00
Total	$29.88	$50.50	$85.00	$47.75	$77.50	$135.50

Total cost of operation per year on basis of operating full capacity four months, and one-quarter capacity fou months, equivalent to full operation for 216 days (60% load factor).

Daily operating expense during period of full operation	$6,454.00	$10,908.00	$18,360.00	$10,314.00	$16,740.00	$29,268.00
All labor expense for balance of year	2,348.00	3,672.00	5,328.00	2,736.00	4,176.00	5,832.00
5% depreciation on cost of mechanical equipment	1,825.00	3,325.00	6,000.00	1,500.00	2,775.00	4,950.00
3% depreciation on building	480.00	1,050.00	1,800.00	480.00	1,050.00	1,800.00
5% on total investment for repairs, taxes, water and incidentals	2,875.00	5,475.00	9.600.00	2,550.00	4,925.00	8,550.00
Total annual expense	$13,982.00	$24,430.00	$41,088.00	$17,570.00	$29,666.00	$50,400.00
Number of tons produced annually	10,800	21,600	43,000	10,800	21,600	43,000
Total cost per ton of ice per annum	$1.30	$1.11	$0.96	$1.62	$1.40	$1.17
Saving of oil engine plants per ton	.32	.33	.21	.....	.....	.....
Yearly saving oil engine plants	$3,588.00	$6,860.00	$9,312.00	.....	.....	.....

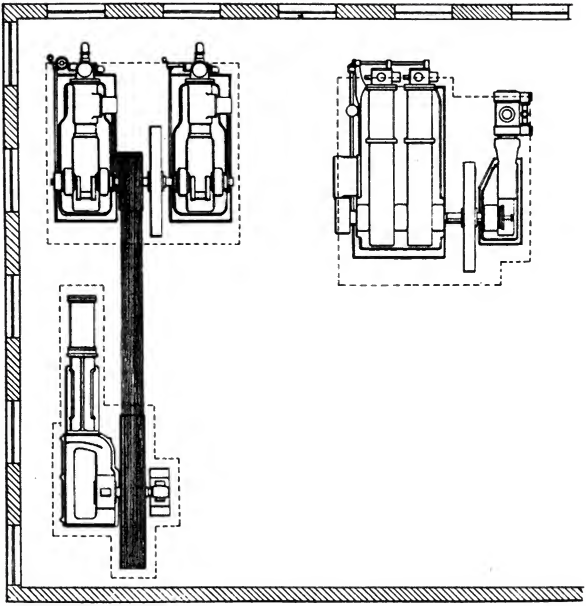


Fig. 16.—Comparison of rope-driven compressors and direct-connected compressors of 100 tons ice-making capacity.

with a unit of the same capacity wherein the oil engine and ammonia compressor are direct connected. There is no question but that a perfect high speed compressor would greatly accelerate the installation of oil-engine-driven ice plants. The following are the benefits to be derived from a direct connected unit:

1. A saving of 60 per cent in floor space, resulting in a reduced cost of building.
2. A reduction in required horsepower of about 15 per cent, due to the elimination of the loss in the transmission and the excessive friction loss in the bearings.
3. A saving in first cost through eliminating the transmission equipment.
4. Reduced wear on the main bearings of engine and compressor.
5. A lighter and cheaper ammonia compressor.

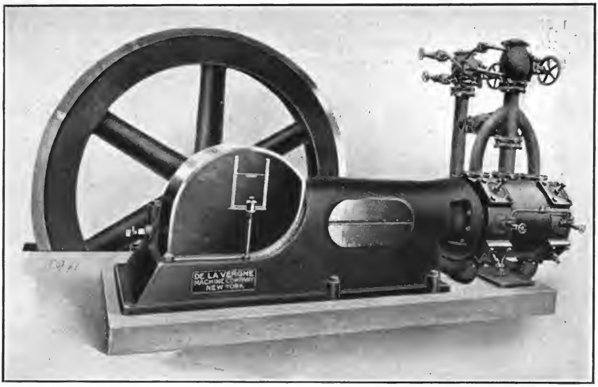


Fig. 17.—De La Vergne compressor of 50 tons ice-making capacity, direct connected to oil engine.

6. A saving in fuel consumption of about 15 per cent.

7. A reduced cost of maintenance on account of the elimination of the transmission equipment.

As far as I know, the only oil-engine-driven, direct connected plant that has been in operation for any length of time is the Alpine Ice Company, of Kansas City, which has been described in *Ice & Refrigeration*. I understand the plant has given satisfaction, but that since starting about six months ago it has been running at the comparatively low speed of 125 revolutions per minute. Later it will probably be speeded up to 160-170 revolutions per minute, which is the normal speed of the engine.

Fig. 17 shows a 140-horsepower De La Vergne type "FH" engine direct connected to a De La Vergne high speed compressor of 15 x 15-inch cylinder dimensions, and of an ice-making capacity of 50 tons. The speed of this unit is 164 revolutions per minute.

As soon as the high speed ammonia compressor has proven a success by a long period of commercial use, there is no question but that it will come into general use, and that the cost of an oil-engine-driven unit will be below that of a steam plant. Of greater importance is the fact that the entire cost of installation, including the buildings, will be considerably less for an oil-engine-driven plant than for a steam plant, and there is no better way to stimulate the introduction of new equipment than to make possible a saving in initial expense.

The future of the oil-engine-driven ice plant is assured. Mistakes have been made in the past and will be made in the future, but every month sees an improvement in oil engine equipment and in compressor design, every month sees a lower production cost for both, and every month shows an increased interest and increased demand for the direct connected oil-engine-driven ice plant.

THE RATE OF HEAT TRANSFER IN DOUBLE-PIPE BRINE COOLERS*

BY FRED OPHULS AND VAN R. H. GREENE, NEW YORK, N. Y.

The determination of the rate of heat transfer in double-pipe brine coolers brings with it the necessity of solving many interesting and varied problems. It is a subject that has not received much consideration at the hands of the manufacturers of this class of apparatus. Many attempts on the part of the De La Vergne Machine Company, of New York City, to design a double-pipe brine cooler which would give a predetermined brine range for a given suction pressure at the ice machine prompted them, in the spring of 1912, to erect a thirty-five ton test plant, in order to look into this and many other allied subjects. It is due to the courtesy of the De La Vergne Machine Company that we are able to submit for your consideration the following data:

The test plant consisted of an 11½ x 20-inch double-acting, horizontal refrigerating machine, direct connected to a 20 x 20-inch Corliss steam engine. The suction side of the refrigerating machine was connected by means of a 3½-inch pipe to four standard double pipe brine coolers, each twelve pipes high and 21 feet long, made of 2-inch and 3-inch pipe. The discharge from the ice machine was connected both to a battery of atmospheric and double pipe ammonia condensers, and the liquid from the receiver was fed into the side of an accumulator connected to the bottom of the brine coolers, so that these coolers could be operated upon the flooded system. The refrigerating load was produced by a secondary warm brine system which passed in turn through a steam heater and exchangers in closed circuit, while the main brine system was pumped, first through the heat exchanger in counter current flow to the warm brine, and then through the double pipe brine coolers, also forming a closed and independent circuit.

By suitable cross connections at the brine coolers, the brine could be made to travel in either parallel or counter current flow with the ammonia. The arrangement of this apparatus is shown diagrammatically in Fig. 1.

While experimenting with the proper height of brine cooler to

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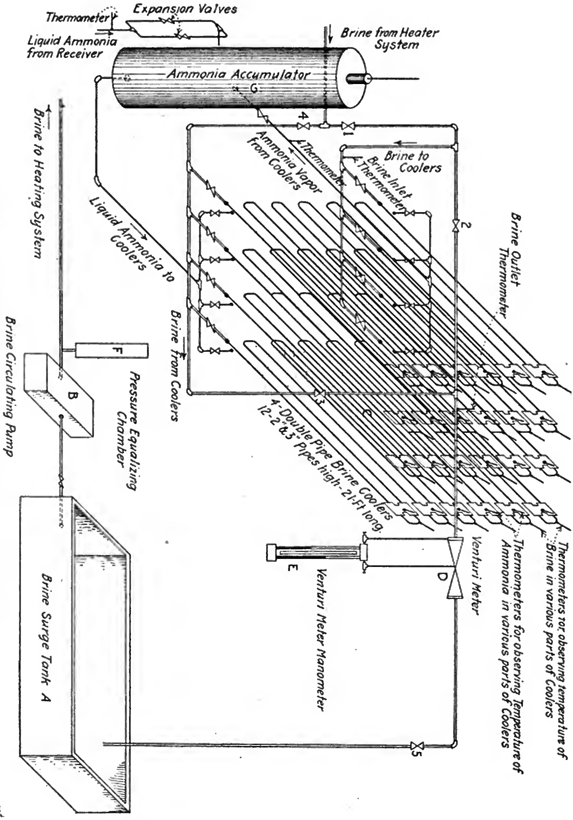

Thermometer
Expansion Valves
Liquid Ammonia from Receiver
Brine from Heater System
Ammonia Accumulator
G
4
1
2
3
5
Thermometer
Ammonia Vapor from Coolers
Brine to Coolers
Brine Inlet Thermometer
Brine Outlet Thermometer
Brine to Heating System
Liquid Ammonia to Coolers
Brine from Coolers
F
Pressure Equalizing Chamber
B
Brine Circulating Pump
C
4-Double Pipe Brine Coolers 12-2"&3" Pipes high-21-Ft long.
Thermometers for observing temperature of Brine in various parts of Coolers
Thermometers for observing Temperature of Ammonia in various parts of Coolers
Venturi Meter
D
E
Venturi Meter Manometer
Brine Surge Tank A

Fig. 1.—Diagrammatic sketch of brine cooler test plant

produce a given result, it did not take long to ascertain that the pressure required to force brine through a double pipe brine cooler became excessive when the number of pipes high were increased; and as this had a decided bearing upon the proper design of a brine cooler, we have decided to touch upon this subject in conjunction with our regular topic.

BRINE FRICTION

In determining the pressure drop through these brine coolers, all tests recorded were made with the brine flowing up through the coolers, and to obtain a good check on the results tests were made with one, two, three and four stands in parallel. From the brine cooler the brine flowed through a 2-inch Venturi meter, which was equipped with a mercury manometer having a scale reading in gallons per minute for brine of 1.2 specific gravity. Both this meter and the manometer were calibrated by the manufacturers, the Builders Iron Foundry Company, of Providence, R. I., and found to be correct. However, this record was also checked by the displacement figures obtained by taking simultaneous readings of the stroke and revolutions of the direct-acting steam brine pump.

To decrease to a minimum the fluctuations in quantity of brine flowing through the system, which made difficult the accurate reading of the Venturi manometer, due to the changes in the speed of the brine pump during each revolution, an air chamber was installed in the discharge pipe from the pump.

Simultaneous readings were taken of all instruments as soon as the pump speed had been adjusted accurately, to the desired quantity of brine flowing through the system.

In Table 1 all the readings and calculations required for an accurate determination of the pressure drop are given.

The higher gauge pressures were recorded by standard low pressure steam gauges, calibrated before the test was begun. and due allowance was made for the location of the gauge, with respect to the center of the brine headers. The low pressures were recorded by an open-end mercury manometer placed with the 0° pressure reading in the same horizontal plane with the center line of the brine cooler headers. In the case of the brine flowing up through the coolers, the pressures in the lower brine header were recorded by a calibrated pressure gauge, and in the upper header by a mercury manometer. The corrections made in the readings were as follows:

Pressure Gauge—Correction for calibration caused by location of gauge in reference to center of brine header.

Mercury Gauge—No corrections required.

Pressure Differences—Correction for height of cooler. For the coolers tested and the specific gravity of the brine used, this correction was 2.59 pounds per square inch, which amount had to be deducted from the pressure difference to obtain the net pressure drop in the coolers for frictional resistance only. The friction head in feet is obtained by multiplying the pressure in pounds per square inch by 1.95.

To obtain uniformity in the results for plotting the values on

TABLE 1.—RESULTS OF TESTS ON FRICTION LOSSES IN THE 2-INCH PIPE COIL OF A STANDARD DOUBLE-PIPE BRINE COOLER, 12 (2-INCH AND 3-INCH) PIPES HIGH, 21 FEET LONG OVERALL.

Venturi Meter Readings, Gallons Per Minute.	Gauge No. 1 Pressures in Lbs. Per Square Inch.		Gauge No. 2 Pressures in Lbs. Per Sq. In.		Pressure Difference in Lbs. Per Sq. In.		Friction Head in Feet.
	As Read.	Corrected.	As Read.	Corrected.	Actual.	Corrected for Cooler Height.	
No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7	No. 8
FOUR BRINE COOLERS IN PARALLEL—BRINE UP THROUGH COOLERS.							
272.5	36	35	8.12	8.12	26.88	24.29	47.3
262.5	34	33.2	7.31	7.31	24.89	22.30	43.5
251	32.5	31.8	7.5	7.5	24.3	21.71	41.2
241	31	30.3	7.25	7.25	22.05	19.46	37.9
228	28	27.5	6.75	6.75	20.75	18.16	35.4
220	27.5	26.8	6.75	6.75	20.05	17.46	34
210	26	25.3	6.67	6.67	18.63	16.04	31.23
201	24	23.5	6.67	6.33	17.17	14.58	28.4
189	22	21.5	6.25	6.25	15.25	12.66	24.62
181.5	20.25	19.75	6.5	6.5	13.25	10.66	20.75
170	19.5	19	7	7	12	9.41	18.32
155	16.5	16.2	5.5	5.5	10.7	8.11	15.8
139.5	15.37	15	6	6	9	6.41	12.5
123.5	14.37	14.13	6.25	6.25	8.05	5.46	10.6
100	11.75	11.5	5.75	5.75	5.75	3.16	6.13
THREE BRINE COOLERS IN PARALLEL—BRINE UP THROUGH COOLERS							
257	44	43	8.25	8.25	34.75	32.16	62.66
251	42.5	41.5	7.75	7.75	33.75	31.16	60.6
237.5	37.5	36.5	6.33	6.33	30.17	27.58	53.75
225	35	34	6.75	6.75	27.25	24.66	48.1
211.5	33	32	7.33	7.33	24.67	22.08	43.1
201	30	29.3	6.67	6.67	22.63	20.04	39
185	29	28.3	6.67	6.67	21.63	19.04	37.1
164.5	23	22.5	6.13	6.13	16.37	13.78	26.85
150	22	21.5	7.25	7.25	14.25	11.66	22.70
131	16.37	16.13	5	5	11.13	8.54	16.64
110	15.37	15.13	6.5	6.5	8.63	6.04	11.78
101	17	17	9	9	8	5.41	10.54
TWO BRINE COOLERS IN PARALLEL—BRINE UP THROUGH COOLERS.							
200	52	51	7.25	7.25	43.75	41.16	80.3
190.5	47	46	6.33	6.33	39.67	37.08	72.3
180	43	42	6.75	6.75	35.25	32.66	63.6
169.5	39	38	6.25	6.25	31.75	29.16	56.8
160	37	36	6.67	6.67	29.33	26.74	52.2
140	30.5	29.8	6	6	23.8	21.21	41.4
124.5	27	26.3	6.5	6.5	19.8	17.21	33.6
99	18	18	5.75	5.75	12.25	9.66	18.82
90	16	16.	5.33	5.33	10.67	8.08.	15.76
ONE BRINE COOLER—BRINE UP THROUGH COOLER							
120	65	63.5	6	6	57.5	54.91	107
113	56	55	5.13	5.13	49.87	47.28	92.1
101	47.5	46.5	4.88	4.88	41.62	39.03	76.2
90	36	35	5	5	30	27.41	53.5

TABLE 2.—SUMMARY.

GALLONS OF BRINE PER MINUTE PUMPED THROUGH EACH STAND AND CORRESPONDING FRICTION HEAD IN FEET.

BRINE FLOWING UP THROUGH COOLERS.

FOR FOUR STAND TESTS.		FOR THREE STAND TESTS.	
Gallons Per Minute Per Stand.	Friction Head in Feet.	Gallons Per Minute Per Stand.	Friction Head in Feet.
68.1	47.3	85.7	62.6
66.8	43.5	83.8	60.7
63.3	41.2	79.1	53.7
60.3	37.9	75.0	48.1
57.0	35.4	70.5	43.1
55.0	34.0	67.0	39.0
52.5	31.2	61.7	37.1
50.3	28.4	54.8	26.9
47.3	24.6	50.0	22.7
45.4	20.8	43.7	16.6
42.5	18.3	36.7	11.8
38.8	15.8	33.7	10.5
34.9	12.5		
30.9	10.6		
25.0	6.2		

FOR TWO STAND TESTS.		FOR ONE STAND TESTS.	
Gallons Per Minute Per Stand.	Friction Head in Feet.	Gallons Per Minute Per Stand.	Friction Head in Feet.
100.0	80.3	120.0	107.0
95.2	72.3	113.0	92.1
90.0	63.6	101.0	76.2
84.7	56.8	90.0	53.5
80.0	52.2		
70.0	41.4		
60.7	33.6		
49.5	18.8		
45.0	15.8		

co-ordinate paper, the amount of brine in gallons per minute, per brine cooler section, and the corresponding friction head in feet, are summarized in Table 2. These values are plotted on logarithmic cross section paper, Plate 1, and the heavy line at an angle of 60° to the horizontal is the mean curve through all the points plotted. On Plate 2 will be found the same points and curve plotted on ordinary cross-section paper.

The equation of this curve is $h_b = .01043\ (G_b)^2$, where h_b = the friction head in feet for brine of 1.888 specific gravity. G_b = gallons of brine per minute pumped through the cooler.

This formula, obtained from the foregoing tables, is applicable only to the coolers tested, and to make the results of these tests applicable to determine the resistance to the flow of liquids through

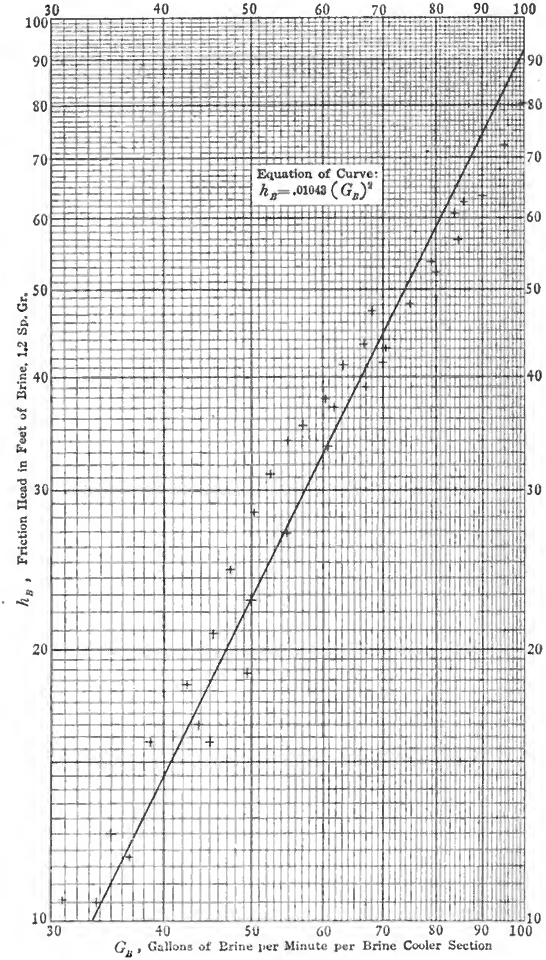


Plate I.—Pressure required to force brine through the 2-in. pipe coil of a standard double-pipe brine cooler. Twelve 2-in. and 3-in. pipes high and 21 ft. long overall.

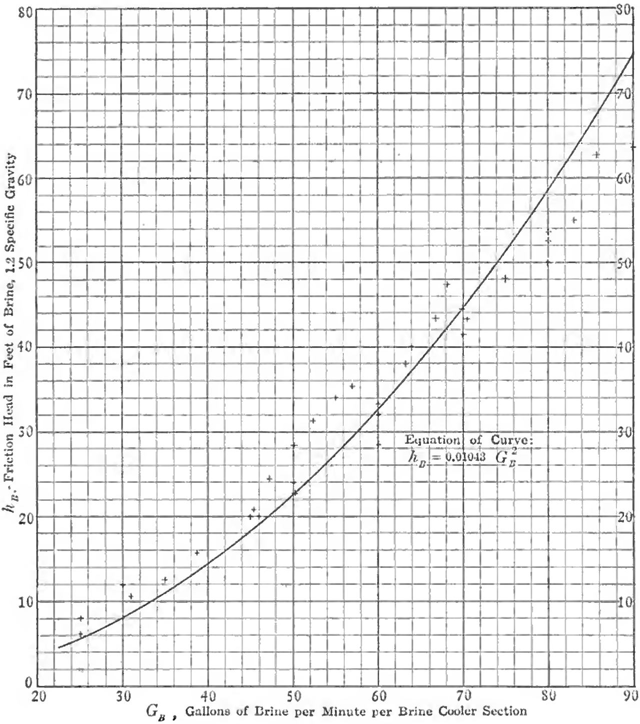


Plate II.—Pressure required to force brine through the 2-in. pipe coil of a standar double-pipe brine cooler. Twelve 2-in. and 3-in. pipes high and 21 feet long overall.

pipe coils of similar construction, a new constant was found for Arthur J. Martins' formula,

$$p = \frac{w^2 L \left(1 + \frac{3.6}{d} \right)}{7{,}000\, D\, d^5} \qquad \text{(Equation 1)}$$

In this formula, p = the friction loss in pounds per square inch, w = pounds of liquid flowing through the pipe per minute,

d = inside diameter of pipe in inches, D = density of fluid in pounds per cubic foot, and L = length of pipe in feet.

If we let f = the friction head in feet and Q = gallons of fluid pumped per minute, Martins' equation can be written

$$Q = 52.15 \sqrt{\frac{fd^5}{L\left(1+\frac{3.6}{d}\right)}} \qquad \text{(Equation 2)}$$

Equation 2 can be used for any fluid in straight lengths of pipe.

To convert this formula for use in connection with flat pipe coils, let us take one point in the curve on Plate 1. That is, for 60.7 gallons of brine circulated per minute, a net pressure drop of 17.2 pounds is obtained, and this is equivalent to a friction head of 33.6 feet. As previously advised, these figures are for a flat pipe coil containing 240 feet of 2-inch pipe (a single brine cooler being 12 pipes high and 20 feet long) together with 11 return bends, having each a mean total length of 10.63 inches or a total of 250 feet of 2-inch pipe per cooler. The inside diameter, d, of a 2-inch pipe is 2.067 inches, and by substituting these values in Equation 2 we find that

$$60.7 = A \sqrt{\frac{33.6 \times (2.067)^5}{250\left(1+\frac{3.6}{2.067}\right)}}$$

By solving this equation we find that A is equal to 51.4, and the new formula will then read

$$Q = 51.4 \sqrt{\frac{fd^5}{L\left(1+\frac{3.6}{d}\right)}} \qquad \text{(Equation 3)}$$

The average amount of slip found for the duplex direct-acting steam pump by comparison with the Venturi manometer readings was 3.5 per cent, confirming the correctness of the readings of this instrument as used in the following calculations, and deductions in connection with the heat transfer in these brine coolers.

HEAT TRANSFER IN GENERAL

The determination of the laws governing heat transfer is very difficult on account of the many variables that enter into the problem. All kinds of fluids are used in many combinations to obtain cooling or heating. The simpler problems are encountered when the heat exchange is carried on between the so-called permanent gases or between liquids or from a permanent gas to a liquid or vice versa. The change of volume for permanent gases or liquids when heated or cooled through moderate ranges of temperature is very small and has little if any bearing on the subject of heat transfer. For such problems a general formula has been found that is applicable in most cases. This formula is:

$$k = a\sqrt{v}. \qquad \text{(Equation 4)}$$

k is the heat transfer coefficient or the rate of heat transfer in B. t. u. per square foot of effective cooling or heating surface per hour per 1° F. mean temperature difference.

a is a constant varying to some extent with the fluids used.

v is the mean relative velocity of the fluids in feet per second.

That the heat transfer coefficient should increase with an increase in the speed of flow of either one or both of the fluids can be well understood when one considers a plain surface maintained at a constant temperature by any source of heat over which flows a stream of colder water. The water will be heated and the water near, or in contact with, the heated surface will be heated more quickly than the water farther away from the surface. So that at any one point of the surface and in a direction perpendicular away from it, the water will be less in temperature the farther it is away from the heated surface. If the speed of flow of the water over the heated surface is increased it follows that the decrease in the temperature of the water at any one point away from the surface, and in a direction perpendicular to it, is greater, and therefore the transfer of heat greater than in the former case. For flat streams of the fluid the increase in the rate of heat flow will be greater than for streams of circular section. Equation 4 was primarily determined from the exchange of heat between condensing steam in closed, trapped pipe coils and air flowing either parallel to or across the coils. In this case the velocity of the air alone effected the value of k, because the increase in the flow of the steam in the trapped pipe coil, for an increase in the amount of heat exchanged between the steam and the air was not sufficient to affect the value of k. If the steam had been flowing through the coil at largely varying speeds, so that not only condensed steam but also vapor flowed from the discharge end of the coil, the rate of heat transfer would have been materially affected by the speed of the steam in the coil as well as by the speed of the air along or across the coil. In that case the form of equation (4) would either have to be changed or else a value for v found that properly took into consideration the speed of both fluids and their effect on the rate of heat flow.

The problem is further complicated by the condition of the cooling or heating surface, whether smooth or rough, or shaped in some way as to mix the fluid passing over it. The more violent the mixing action the greater will be the rate at which the heat is transferred, irrespective of the mean temperature difference.

When one of the fluids is a vapor which is being condensed or a liquid being evaporated, the determination of the law of heat transfer becomes still more difficult. The reason is that for a condensing vapor the speed of flow decreases and for an evaporating liquid the speed of flow of the vapor formed increases rapidly, espe-

TABLE 3.—AVERAGE READINGS OF ALL TESTS MADE ON DOUBLE-PIPE BRINE COOLERS AND TOTAL TONNAGE DEVELOPED.

DATE, 1912.	Number of Test.	Number of Brine Coolers in Use.	Quantities.		Temperatures in Degrees Fahrenheit.										Pressures, lbs. per sq. in.		Tonnage.		
					Brine.			Brine Cooler Room.			Ammonia.								
			Gallons of Brine per Minute.	Gallons of Brine per Minute per Stand.	Into Coolers.	Out of Coolers.	Range.	Front.	Back.	Mean.	Suction at Machine.	Suction into Accumulator.	Before Exp. Value	Discharge at Machine.	Ammonia Discharge.	Ammonia Suction, Mercury Column.	By Brine.	Heat Leakage.	Total.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Aug. 26	204	2	100	50	25.1	19.7	5.4	51.1	51	51	19.8	.....	82.6	264	193.6	22.8	18.90	0.08	18.98
Aug. 26	205	2	100	50	25.77	20.15	5.62	49.6	45.3	47.45	9.03	21.4	87.53	232	188.3	22.7	19.67	0.09	19.76
Aug. 28	210	2	100	50	20.84	16.1	4.74	3.73	38.16	37.73	11.3	19.46	75.3	245	174.16	17	16.59	0.09	16.68
Aug. 28	211	2	100	50	18.08	13.47	4.61	32.93	34.3	33.61	5.6	8	79.7	229	165.2	18.43	16.14	0.10	16.24
Aug. 28	212	2	100	50	17.94	13.3	4.64	34.	32.5	33.25	4.5	7.2	78.6	239.6	162.8	18.55	16.24	0.10	16.34
Sept. 9	601	3	150	50	25	20.71	4.29	34.1	34.9	34.5	5.8	32.7	87.5	175	170	20.9	22.52	0.12	22.64
Sept. 9	602	4	150	37.5	24.2	19.3	4.8	32.9	32.6	32.7	8.7	14.1	87.9	185	172	22.3	25.2	0.12	25.32
Sept. 9	603	4	150	37.5	19.4	14.5	4.9	29.	29.4	29	19.7	26.2	87.5	264	169	17	25.72	0.13	25.85
Sept. 10	501	4	150	37.5	13.36	9.05	4.31	18.4	19.3	80.1	25	3.6	91.1	300	182	15.64	22.62	0.15	22.77
Sept. 11	502	4	150	37.5	13.69	9.68	4.01	35	16	25.5	9.2	6	80.	105	155	20	21.05	0.13	21.18
Sept. 12	503	4	180	45.	11.84	8.11	3.73	19.9	22.4	21.1	—0.3	2.6	84.1	248	198	15.9	23.5	0.12	23.62
Sept. 13	504	4	125	31.25	12.7	7.6	5.1	24.1	23.4	23.75	7.7	2.8	94.4	305	211	15.51	22.36	0.13	22.49
Sept. 16	505	4	100	25	32	23	9	30.1	31	30.5	9.6	19.7	87.7	224	185	22.7	31.5	0.11	31.61
Sept. 16	604	4	100	25	38.8	28.8	10	32.4	32.8	32.6	12.5	33.9	88	235	197	24.6	35	0.11	35.11
Sept. 17	506	4	260	65	28.21	24.71	3.5	29	28.2	28.7	25.9	.....	84.5	212.5	187	24.6	32.4	0.12	32.52
Sept. 17	507	4	260	65	33.08	29.48	3.6	34	32.8	33.4	39.2	.....	80.3	270	178	24.08	32.76	0.11	32.87
Sept. 17	508	4	260	65	32.81	29.57	3.24	32.2	33.1	32.3	18.1	.....	80	......	174	29.5	29.48	0.10	29.58
Sept. 18	509	4	250	62.5	24.75	21.82	2.93	27.7	29.3	28.5	11.5	.....	83.6	165.6	177	.....	25.63	0.11	25.74

Sept. 19	510	4	250	62.5	24.94	21.27	3.67	26.3	28.6	27.45	9.7	.....	80.2	234.8	185.4	21.74	32	0.13	32.13
Sept. 19	511	4	250	62.5	27.47	23.92	3.55	26.83	29.83	28.3	10.44	.....	81.9	239	189.3	23.3	31.2	0.13	31.33
Oct. 7	301	4	175	43.75	12.18	8.53	3.65	21.1	21.4	21.2	31.2	16.5	90.9	292.6	190	16.22	22.35	0.15	22.50
Oct. 8	302	3	180	60	16.27	13.46	2.81	22.3	21.5	22	18.67	4.94	86.6	274	199	17.82	17.83	0.12	17.95
Oct. 8	303	3	180	60	12.02	9.03	2.99	20	18.6	19.2	23.3	1.5	80.5	264	189	15.64	18.83	0.11	18.94
Oct. 9	605	3	180	60	18.7	14.9	3.8	23	24.9	23.6	3.5	14.2	83	269	194	13.6	23.94	0.11	24.05
Oct. 9	512	3	180	60	25.85	22.28	3.66	35.5	32.5	34	11.96	33.6	84.2	288	184	14.1	23.05	0.09	23.14
Oct. 10	606	4	180	45	19.9	16	3.9	26.9	30.4	.....	24.4	32.8	86.5	290	187	15.5	24.57	0.13	24.7
Oct. 10	513	4	180	45	14.8	10.73	4.07	19.7	24.4	22	4	8.7	89	278	181	15.14	25.64	0.14	25.78
Oct. 11	515	4	180	45	14.07	9.75	4.32	18.7	17.65	18.18	—4.7	13.4	76	253	172.6	12.8	27.21	0.12	27.33
Oct. 11	516	4	180	45	16.38	11.99	4.39	19.4	18.8	19.1	—2	15.3	83.7	232.6	177	14.4	27.65	0.12	27.77
Oct. 16	408	3	130	43.33	16.04	11.88	4.16	20.4	22.6	21.4	2.62	.....	81.8	164	212	16	18.95	0.10	19.05
Oct. 18	403	2	130	65	19.10	14.98	4.12	31	31.2	31.1	11.9	.....	83.3	270	212.5	17.3	18.74	0.10	18.84
Oct. 23	409	2	130	65	14.18	10.94	3.24	35	35.2	35.1	—2.5	7.9	77.2	186	175.5	14.25	14.75	0.09	14.84
Oct. 24	404	3	130	43.33	11.97	7.92	4.05	20.4	23.7	22	7.8	—4	75.5	274	185.5	14.1	18.42	0.10	18.52
Oct. 24	406	3	130	43.33	13.10	8.41	4.69	23.2	27.2	24.9	4.2	+0.02	72.8	239	179	14.9	21.3	0.09	21.39
Oct. 25	304	3	130	43.33	11.86	7.74	4.12	18	21.9	20.5	6.1	0.47	73.6	268	179	14	18.84	0.09	18.93
Oct. 28	407	3	130	43.33	11.72	6.53	5.19	20.7	24	22.3	19	.05	78.6	276	173.5	13.25	23.6	0.11	23.71
Oct. 28	405	3	130	43.33	11.90	6.86	5.04	18.2	22.25	20.7	19.2	1.9	81.7	279.4	178	13.5	22.93	0.12	23.05

cially if the change of state is carried on in a pipe coil of uniform cross-section. This decrease and increase, respectively, of the vapor speed tend to effect the rate of heat transfer in the former case, as any speed change that will tend to remove the condensed vapor more quickly from the cooling surface will materially increase the value of k, irrespective of the condensing fluid, while in the latter case any speed change that will result in a better wetting of the heating surface will have the same effect on the value of k. The double-pipe brine coolers on which the tests were made that form the subject of this paper are of the latter type of heat interchangers.

THE RATE OF HEAT TRANSFER IN DOUBLE-PIPE BRINE COOLERS

In order to accurately determine the heat transfer, thermometers were inserted in each one of the return bends at the rear end of the brine coolers, as well as in the annular space wherein the ammonia traveled. These readings enabled us to determine how much work was being done in each pipe, and showed us at once whether the gas was saturated throughout the cooler length or was being superheated in the upper sections.

The apparatus used, except the refrigerating machine, its appurtenances and the brine heating system, is shown diagrammatically in Fig. 1 and is briefly described in the introduction.

Table 3 gives the average readings for all the tests made to determine the rate of heat transfer in double-pipe brine coolers, and in column 20 the total tonnage developed in each case. In Table 4 will be found, column 25, the average mean value of k for each of the tests made. For determining k the following equation was used:

$$H = S\,k\,t_a \qquad \text{(Equation 5)}$$

H is the amount of heat removed by the evaporation of liquid ammonia in the brine coolers, B. t. u. per hour per brine cooler section.

S, the effective cooling surface in each cooler section = 149.2 square feet.

t_a is the logarithmic mean temperature difference between the brine and the ammonia in the cooler, degrees F.

k is the mean value of the heat transfer coefficient, in B. t. u. per square foot per hour per 1° F. mean temperature difference.

In finding the true mean temperature difference between the brine and the liquid ammonia in the coolers, due consideration was given to the difference between the pressure in the suction pipe at the refrigerating machine and that existing at the liquid inlet to and the vapor discharge end of the coolers and the necessary corrections made. The temperatures given in column 22, Table 4, are the mean temperatures of the evaporating liquid in the coolers.

Before attempting to use the values of k found to get some idea of the manner in which it changes with varying conditions of opera-

TABLE 4.

DETERMINATION OF THE MEAN HEAT TRANSFER COEFFICIENT AND THE SPEED OF THE BRINE IN THE COOLERS

No. of Test. 2	Heat Removed in Brine Coolers B.t.u. per Hour. 21	Mean.Temp. Temp. of Liquid Ammonia in Coolers, °F. 22	t_d Logarithmic Mean Temp. Diff. Between Brine and Ammonia. °F. 23	H Heat Removed per Brine Cooler Section B.t.u. per Hour. 24	k Heat Transfer Coefficient B.t.u. per sq. Foot per Hour per 1 °F. 25	v_b. Velocity of Brine Through Coolers, fee per Second. 26
204	227,760	10.8	11.44	113,880	66.7	4.78
205	237,120	9.3	13.51	118,560	58.8	4.78
210	200,160	3.9	14.50	100,080	46.3	4.78
221	194,880	5.7	9.96	97,440	66.1	4.78
212	196,080	5.8	9.69	98,040	67.8	4.78
601	271,680	8.7	14.08	90,560	43.1	4.78
602	303,840	10.2	11.42	75,960	44.6	3.58
603	310,200	3.9	12.94	77,550	40.1	3.58
501	273,240	2.0	9.09	68,310	50.3	3.58
502	254,160	5.7	5.80	63,540	73.4	3.58
503	283,440	2.4	7.46	70,860	63.6	4.31
504	269,880	1.9	8.04	67,470	56.2	2.99
505	379,320	10.7	16.50	94,830	38.5	2.39
604	421,320	12.8	20.70	105,330	34.1	2.39
506	390,840	12.7	13.72	97,710	47.7	6.22
507	394,440	12.2	18.80	93,610	33.4	6.22
508	353,960	17.8	13.34	88,490	44.4	6.22
509	308,880	8.9	14.34	77,220	36.1	5.98
510	385,560	9.6	13.47	96,390	48.0	5.98
511	375,960	11.3	14.37	93,990	43.8	5.98
301	270,000	2.8	7.45	67,500	60.7	4.19
302	215,400	4.9	9.92	71,800	48.5	5.74
303	217,280	2.0	8.45	72,427	57.4	5.74
605	288,600	0.5	15.53	96,200	41.5	5.74
512	277,680	0.1	24.00	92,560	25.8	5.74
606	296,400	1.8	16.1	74,100	30.8	4.31
513	309,360	1.4	11.32	77,340	45.8	4.31
515	327,960	—1.7	13.82	81,990	39.8	4.31
516	333,240	0.5	13.60	83,310	41.0	4.31
408	228,600	2.6	11.28	76,200	45.2	4.14
403	226,080	4.3	12.67	113,040	59.9	6.22
409	177,840	0.3	12.20	88,920	48.8	6.22
404	222,240	0.1	9.73	74,080	51.0	4.14
406	256,680	1.2	9.41	85,560	60.9	4.14
304	227,160	0.0	9.69	75,720	52.4	4.14
407	284,520	—1.2	10.17	94,840	62.5	4.14
405	276,600	—0.7	9.92	92,200	62.2	4.14
410	286,320	—1.6	10.34	95,440	61.8	4.14
305	277,800	—0.15	9.39	92,600	66.0	4.14
607	478,320	6.7	25.00	159,440	42.8	4.14

tion or various combinations of coolers, it was necessary to investigate the probable accuracy of the values found.

Three important points necessarily enter into this consideration, and they are: Duration of Test, Tonnage per Stand and Quality of the Ammonia Vapor at the discharge from the brine coolers, by which we mean whether the gas was Saturated or Superheated. Discrepancies in one or more of these three points forced us to eliminate the following test runs: 601, 603, 505, 604, 507, 509, 301, 302, 605, 512, 606, 515, 607.

The usual method employed in determining the law of the change in the rate of heat transfer in double-pipe brine coollers is to compare the brine velocity alone with the heat transfer coefficient. The fallacy of this method is shown by the accompanying Plate 3, from which you. can observe how impossible it is to locate any definite clrve through the points plotted.

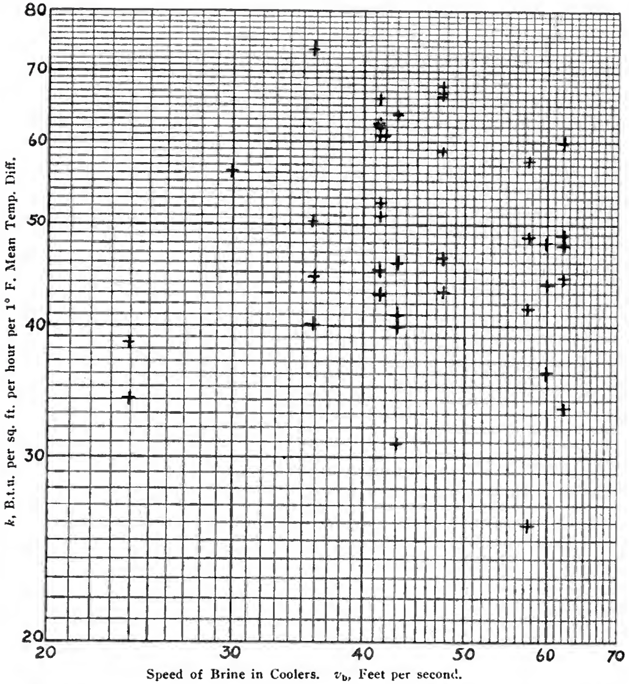


Plate III.—Heat transfer coefficient compared to speed of brine in cooler.

It is a well-known fact, as before stated, that the velocity of flow of fluids when either being liquefied or vaporized seems to effect the rate of heat transfer, so it occurred to us to plot the rate of heat transfer against the speed of the ammonia vapor at the discharge end from the brine coolers instead of the speed of the brine.

In Table 5 the method of determining the speed of the ammonia vapor discharging from the annular space between the 3-inch and 2-inch pipes of the brine coolers will be found. This speed, v_a, is given in column 33. Plate 5 shows the plot of v_a against the respective values of k. The points obtained, while showing an improvement over those obtained on Plate 3, do not show a sufficient tendency to gather about a curve of the general form $k = a\sqrt{v}$ to allow of forming any conclusions from this plot.

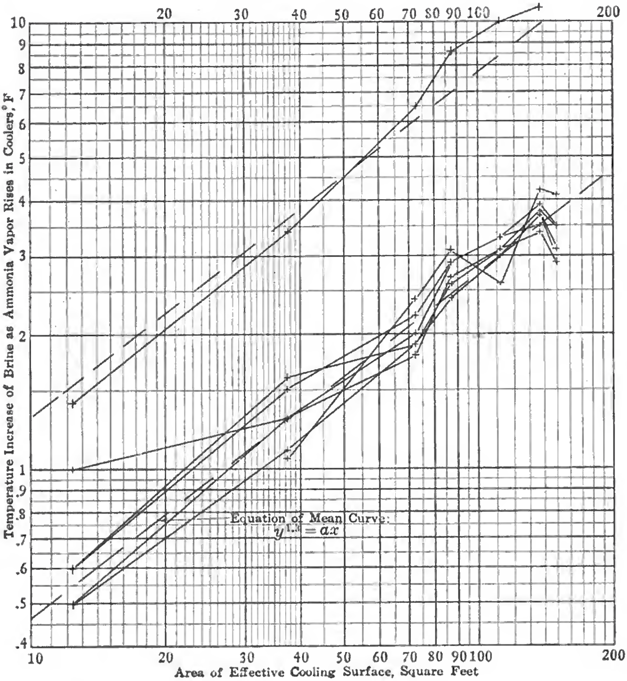


Plate IV.—Temperature increase of brine in cooler as ammonia vapor passes through cooler.

v_a is the final speed of the ammonia vapor in the coolers and it seems reasonable to assume that, if the speed of the vapor forms part of the general equation, the mean speed of the vapor in the coolers must be found, or, better, that speed of the vapor which results in combination with the brine speed v_b in the true mean value of k as recorded in Table 4, column 25.

To obtain this mean speed of the vapor the record of the brine temperatures in the rear end return bends of the coolers is available. As the amount of brine circulated at any one time in each cooler is constant, the temperature increase of the brine from the bottom pipe up is proportional to the amount of liquid evaporated and therefore

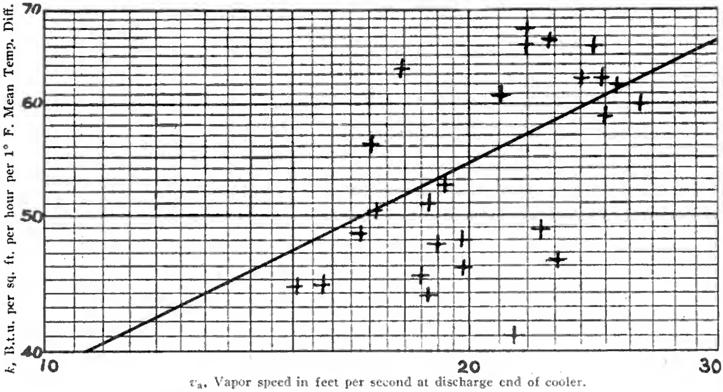


Plate V.—Heat transfer coefficient compared to velocity of ammonia vapor leaving cooler.

to the increase in vapor speed. With this in mind, the points on Plate 4 were plotted and the horizontal distances being the logarithms of the effective cooling surface passed over by the vapor and the vertical distances the logarithms of the brine temperature increase. The general direction of the lines joining the points so obtained can be expressed by the formula

$$y^{1.3} = a\ x; \qquad \text{(Equation 6)}$$

in which y is the total temperature increase of the brine or the vapor speed in feet per second for the effecting cooling surface, x, passed over; a is a constant giving the location of the curve in reference to the X and Y axis. By integration it is found that the arithmetical mean speed of the vapor in the annular space based on Equation 6 is 0.565 v_a.

The values of 0.565 v_a are given in column 34, Table 6, as well as the values of the constant a for all the tests made. On Plate 6 will be found a plot of the values 0.565 $v_a + v_b$, column 35, Table 6, against the values of k, column 25, Table 4. The points of the tests before discarded on account of discrepancies in manner of operating the coolers are left out on this plate. The average value of the constant a was found to be 14.2, so that the average curve for all the points shown would have the following equation:

$$k = 14.2\sqrt{0.565\ v_a + v_b}. \qquad \text{(Equation 7)}$$

The results of this plot are still far from satisfactory, for the

TABLE 5.
CALCULATIONS TO DETERMINE v_a

No. of Test. 2	Weight of Liquid Ammonia Flowing Through Each Cooler, Pounds per Minute. 27	Mean Pressure in Coolers. Pounds per Square Inch Absolute. 28	Volume of Saturated Vapor at Mean Cooler Pressure, Cubic Feet per Pound. 29	Temperature of Vapor Leaving Coolers, °F. 30	Volume of Vapor Corrected for Superheat, Cubic Feet per Pound. 31	Vapor Discharged from Each Cooler, Cubic Feet per Minute. 32	Velocity of Vapor Leaving Cooler, Feet per Second. v_a 33
204	3.9	38.7	7.21	....	7.21	28.1	22.8
205	4.13	37.4	7.45	....	7.45	30.8	25.0
210	3.35	32.9	8.42	19.5°	8.53	28.6	23.2
211	3.33	34.3	8.08	8.0°	8.12	27.1	22.0
212	3.35	34.4	8.06	7.2°	8.06	27.0	22.0
601	3.08	36.8	7.56	32.7°	7.95	24.5	19.9
602	2.62	38.3	7.30	14.1°	7.36	19.3	15.7
603	2.65	32.9	8.42	26.2°	8.85	23.4	19.0
501	2.41	31.5	8.77	3.6°	8.79	21.2	17.2
502	2.18	34.3	8.08	....	8.08	17.6	14.3
503	2.46	31.8	8.95	....	8.95	22.0	17.9
504	2.40	31.4	8.80	....	8.80	21.1	17.1
505	3.22	38.6	7.23	19.7°	7.36	23.7	19.3
604	3.58	40.5	6.906	33.9°	7.21	25.8	21.0
506	3.38	40.5	6.906	....	6.91	23.4	19.0
507	3.35	40.0	6.989	....	6.99	23.4	19.0
508	3.02	45.4	6.188	....	6.19	18.7	15.2
509	2.74	37.0	7.52	....	7.52	20.6	16.7
510	3.30	37.6	7.41	....	7.41	24.4	19.8
511	3.23	39.2	7.126	....	7.13	23.0	18.7
301	2.35	32.1	8.61	16.5°	8.87	20.8	16.9
302	2.50	33.7	8.22	4.9°	8.22	20.6	16.7
303	2.63	31.5	8.77	1.5°	8.77	23.1	18.8
605	3.28	30.3	9.09	14.2°	9.36	30.7	25.0
512	3.12	30.0	9.17	33.6°	9.84	30.7	25.0
606	2.53	31.4	8.80	32.8°	9.39	23.8	19.3
513	2.7	31.0	8.90	8.7°	9.03	24.4	19.8
515	2.75	28.7	9.56	13.4°	9.88	27.2	22.1
516	2.85	30.3	9.09	15.3°	9.25	26.4	21.4
408	2.63	31.9	8.67	....	8.67	22.8	18.5
403	3.91	33.2	8.34	....	8.34	32.6	26.5
409	3.02	30.2	9.12	7.9°	9.12	27.5	22.4
404	2.51	30.0	9.17	—4.0°	9.17	23.0	18.7
406	2.9	30.8	8.95	0.2°	8.95	26.0	21.1
304	2.57	29.9	9.20	0.47°	9.20	23.6	19.2
407	3.25	29.2	9.41	—0.5°	9.41	30.6	24.8
405	3.17	29.4	9.35	1.9°	9.35	29.6	24.1
410	3.24	28.8	9.53	3.7°	9.64	31.2	25.4
305	3.18	29.8	9.23	0.0°	9.25	30.3	24.6
607	5.43	35.2	7.89	16.5°	8.06	43.8	35.6

points found are not grouped closely enough around the average curve to warrant the conclusion that the general Equation 4 is applicable in this case, and that the arithmetical mean speed of the vapor, 0.565 v_a, is the correct one to use to satisfy the equation. Another bad feature of the tests made is that they do not give results covering a wide enough range of speeds and coefficients k to be able to fix the path of the mean curve with any degree of certainty. The reason for this was the construction of the liquid ammonia feed system. An ammonia accumulator located at any fixed height in reference to the brine coolers is not satisfactory for obtaining a wide range of tonnage and still obtain a saturated vapor at the discharge end from the coolers. Hand-operated expansion valves, one for each cooler, would have served the purpose for which these tests were made far better.

TABLE 6.

CALCULATION TO DETERMINE THE VALUE OF CONSTANT *a*, BASED ON THE ARITHMETIC MEAN SPEED OF THE VAPOR PLUS THE BRINE SPEED

No. of Test. 2	$0.565\ v_a$ Feet per Second. 34	$0.565\ v_a+v_b$ Feet per Second. 35	Square Root of $0.565\ v_a+v_b$. 36	Constant *a*. 37
204	12.89	17.67	4.2	15.9
205	14.13	18.91	4.34	13.5
210	13.11	17.89	4.22	10.9
211	12.42	17.20	4.15	15.9
212	12.42	17.20	4.15	16.3
601	11.25	16.03	4.00	10.8
602	8.87	12.45	3.53	12.6
603	10.73	14.31	3.78	10.6
501	9.72	13.30	3.64	13.8
502	8.08	11.66	3.42	21.4
503	10.12	14.43	3.79	16.7
504	9.66	12.65	3.56	15.8
505	10.9	13.29	3.65	10.5
604	11.87	14.26	3.77	9.05
506	10.73	16.95	4.10	11.6
507	10.73	16.95	4.10	8.15
508	8.59	14.81	3.84	11.5
509	9.43	15.41	3.92	9.2
510	11.19	17.17	4.13	11.6
511	10.56	16.54	4.06	10.8
301	9.55	13.74	3.70	16.4
302	9.43	15.17	3 84	12.6
303	10.62	16.36	4.03	14.2
605	14.13	19.87	4.45	9.4
512	14.13	19.87	4.45	5.7
606	10.9	15.20	3.89	7.9
513	11.19	15.50	3.94	11.6
515	12.49	16.80	4.10	9.7
516	12.19	16.50	4.05	10.1
408	10.46	14.60	3.82	11.8
403	14.97	21.19	4.60	13.0
409	12.66	18.88	4.34	11.2
404	10.56	14.70	3.83	13.3
406	11.92	16.06	4.01	15.2
304	10.84	14.98	3.84	13.6
407	14.00	18.14	4.25	14.7
405	13.62	17.76	4.20	14.8
410	14.36	18.50	4.30	14.4
305	13.80	17.94	4.23	15.6
607	20.13	24 27	4.91	8.7

Average value of *a*, leaving out the tests heretofore mentioned, 14.2.

Another attempt was made to see whether the test results available did not afford sufficient data to fix the law governing the rate of heat transfer. It was argued, if the fundamental equation 4 is applicable in this case, that the mean speed of the vapor is combination with the brine speed, resulting in the mean value of *k* found, would be that speed which would correspond to the mean square root speed determined from Equation 6.

The curve of the square root speeds in reference to the effective cooling surface would be:

$$y_1^{2.6} = a\ x. \qquad \text{(Equation 8)}$$

By integration between limits the mean square root speed of the vapor is found to be

$$\frac{1.3}{1.8}\ v_a^{1/2},$$

and the vapor speed corresponding to it will be

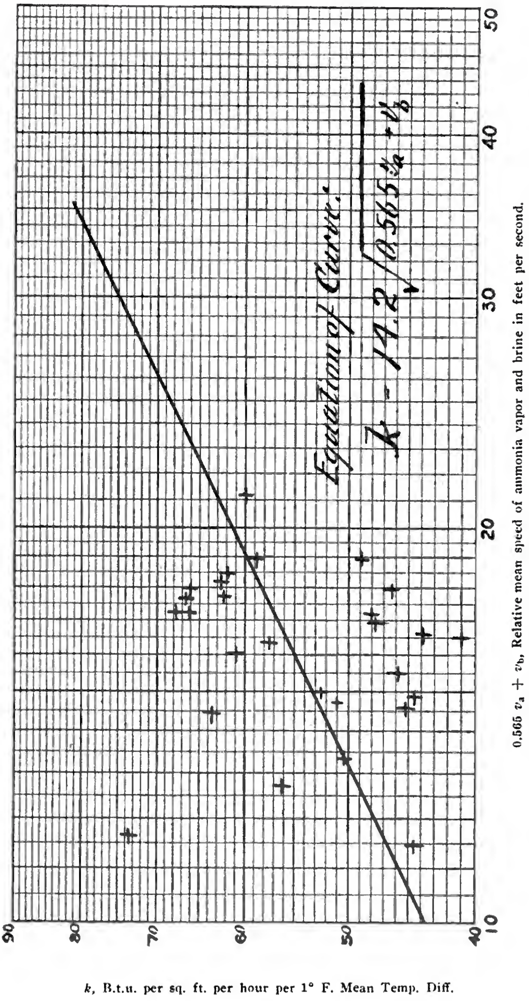


Plate VI.—Heat transfer coefficient compared to relative mean speed of ammonia vapor and brine

TABLE 7.
CALCULATIONS TO DETERMINE THE VALUE OF CONSTANT *b*, BASED ON THE MEAN SQUARE ROOT SPEED OF THE VAPOR PLUS THE BRINE SPEED

No. of Test. 2	$0.521\, v_a$, Feet per Second. 38	$0.521\, v_a + v_b$, Feet per Second. 39	Square Root of $0.521\, v_a + v_b$ 40	Constant *b*. 41
204	11.87	16.65	4.08	16.10
205	13.02	17.80	4.22	13.93
210	12.08	16.86	4.11	11.26
211	11.46	16.24	4.03	16.40
212	11.46	16.24	4.03	16.82
601	10.36	15.14	3.89	11.09
602	8.17	11.75	3.42	13.04
603	9.89	13.47	3.67	10.92
501	8.96	12.54	3.54	14.20
502	7.45	11.03	3.32	22.10
503	9.32	13.63	3.69	17.20
504	8.90	11.89	3.44	16.33
505	10.05	12.44	3.52	10.94
604	10.94	13.33	3.65	9.34
506	9.89	13.47	3.67	13.00
507	9.89	13.47	3.67	9.10
508	7.91	14.13	3.75	11.84
509	8.70	14.68	3.83	9.42
510	10.31	16.29	4.04	11.88
511	9.74	15.72	3.96	11.07
301	8.80	12.99	3.60	16.87
302	8.70	14.68	3.83	12.66
303	9.79	15.53	3.94	14.58
605	13.02	18.76	4.33	9.58
512	13.02	18.76	4.33	5.95
606	10.05	14.36	3.78	8.15
513	10.31	14.62	3.82	12.00
515	11.51	15.82	3.97	10.02
516	11.14	15.45	3.93	10.43
408	9.63	13.77	3.71	12.18
403	13.80	14.02	3.74	16.01
409	11.67	17.89	4.23	11.53
404	9.74	13.88	3.72	13.70
406	10.99	15.13	3.88	15.69
304	10.00	14.14	3.76	13.93
407	12.92	17.06	4.13	15.13
405	12.55	16.69	4.08	15.24
410	13.23	17.37	4.16	14.98
305	12.81	16.95	4.12	15.77
697	18.54	22.68	4.76	8.98

Average value of *b*, leaving out the tests heretofore mentioned, 14.30.

$$\left(\frac{1.3}{1.8}\, v_a^{1/2}\right) = 0.521\, v_a,$$

which differs in this case but slightly from the arithmetical mean speed found from equation 6. For this reason the result of plotting the value $0.521\, v'_a + v_b$, given in column 39, Table 7, against the corresponding values of *k*, column 25, Table 4, as shown on Plate 7, using logarithmic cross-section paper, and on Plate 8, using ordinary cross-section paper, did not prove any more satisfactory. The average value of *b* was found to be 14.3, and the equation of the mean curve:

$$k = 14.3 \sqrt{0.521\, v_a + v_b}. \qquad \text{(Equation 9)}$$

The above would have left the subject in a rather unsatisfactory state if another manner of analyzing the test results obtained had not been available.

On further reflection we found that, in conjunction with the tonnage obtained for each cooler section, the true mean temperature

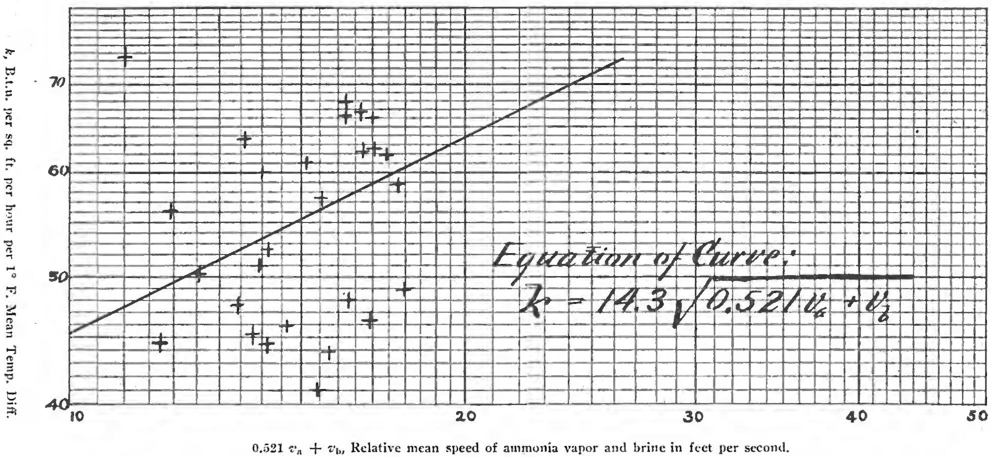


Plate VII.—Heat transfer coefficient compared to relative mean speed of ammonia vapor and brine

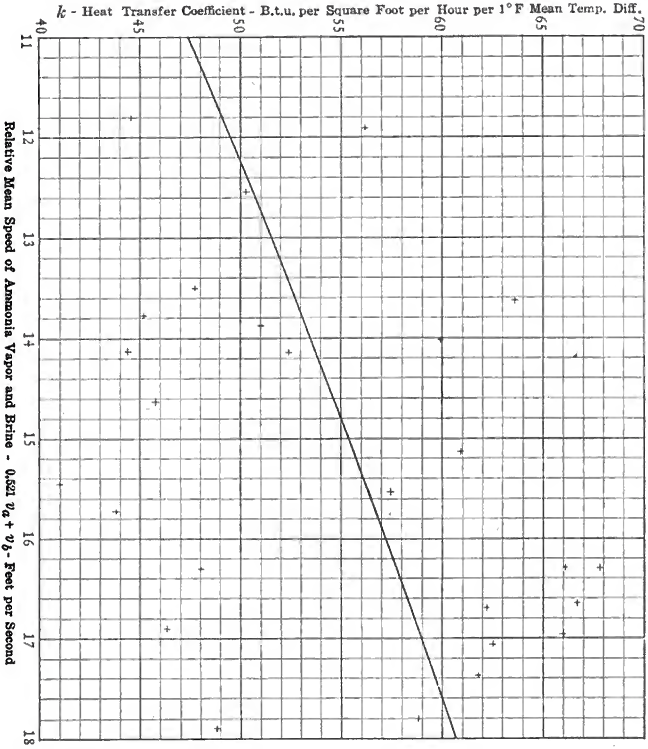
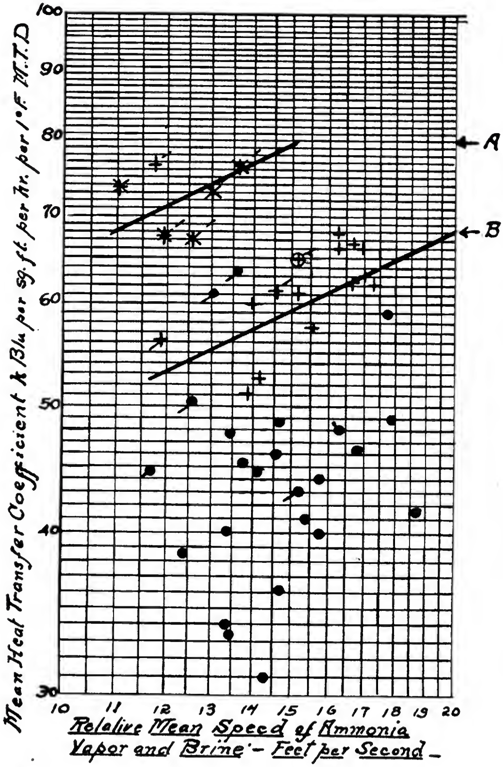


Plate VIII.—Heat transfer coefficient compared to relative mean speed of ammonia vapor and brine

difference between the brine and vaporizing liquid in the cooler was most important for judging the value of the test results obtained. For this purpose Table 8 and Plate 9 are offered for your consideration.

In Table 8 the tonnages per cooler sections are arranged in separate columns between unit limits from 5 tons up and the value of t_d placed next to the tonnage figures. It stands to reason that for those tests, were the value of t_b, is low for the tonnage obtained, the



Note:– ✱ and + Points, Actual Test Results
✱′ and +′ Points, Recalculated "
● and ● Points. Test Results not used because of super-heated vapor in one or more cooler pipes.

Curve A—$k = 20.6\sqrt{0.521\,v_a + v_b}$. 5 to 6 tons refrigeration per cooler section and about 6° F. mean temp. diff.

Curve B—$k = 15.2\sqrt{0.521\,v_a + v_b}$. 6 to 8 tons refrigeration per cooler section and about 10° F. mean temp. diff.

Plate IX—Heat transfer coefficient compared to relative mean speed of ammonia vapor and brine. Some test results corrected for superheated vapor in one or more of the top pipes of the coolers and those test results discarded that could not be so corrected.

TABLE 8.

Comparison of Total Tonnage Obtained for Each Brine Cooler Section and Mean Temperature Difference between Brine and Liquid Ammonia in Cooler.

No. of Test. 2.	TONNAGE DIVISION.												Tests Discarded on Account of Excessive t_d Showing Superheating of Vapor in Top Pipes for Which Corrections Could Not be Made. 54	Number of Pipes that Were Found to be Effective from Temp. Readings in Ammonia Return Bends at Rear of Coolers. 55	k. Heat Transfer Coefficients Used for Plotting Acceptable Points on Plate 9. 56
	13 to 14		9 to 10		8 to 9		7 to 8		6 to 7		5 to 6				
	Actual Tonnage. 42	t_d 43	Actual Tonnage. 44	t_d 45	Actual Tonnage. 46	t_d 47	Actual Tonnage. 48	t_d 49	Actual Tonnage. 50	t_d 51	Actual Tonnage. 52	t_d 53			
204	.....	.....	9.49	11.44	.....	.....	.....	.....	.....	.....	.....	.....	....	....	66.7
205	.....	.....	9.88	13.51	.....	.....	.....	.....	.....	.....	.....	.....	———	....	.....
210	.....	.....	.....	.....	8.32	14.50	.....	.....	.....	.....	.....	.....	———	....	.....
211	.....	.....	.....	.....	8.12	9.96	.....	.....	.....	.....	.....	.....	....	....	66.1
212	.....	.....	.....	.....	8.17	9.69	.....	.....	.....	.....	.....	.....	....	....	67.8
601	.....	.....	.....	.....	.....	.....	7.55	14.08	.....	.....	.....	.....	———	....	....
602	.....	.....	.....	.....	.....	.....	.....	.....	6.33	11.42	.....	.....	———	....	....
603	.....	.....	.....	.....	.....	.....	.....	.....	6.46	12.94	.....	.....	———	....	....
501	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	5.69	9.09	Corrected for superheat	9 pipes...	67.1
502	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	5.29	5.80	....	....	73.4
503	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	5.91	7.46	Corrected for superheat	10 pipes...	76.3
504	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	5.62	8.14	Corrected for superheat	10 pipes..	67.5
505	.....	.....	.....	.....	.....	.....	7.90	16.50	.....	.....	.....	.....	———	....	....
604	.....	.....	.....	.....	8.78	20.7	.....	.....	.....	.....	.....	.....	———	....	....
506	.....	.....	.....	.....	8.13	13.7	.....	.....	.....	.....	.....	.....	———	....	....
507	.....	.....	.....	.....	8.22	18.8	.....	.....	.....	.....	.....	.....	———	....	....
508	.....	.....	.....	.....	.....	.....	7.39	13.34	.....	.....	.....	.....	———	....	....
509	.....	.....	.....	.....	.....	.....	.....	.....	6.43	14.34	.....	.....	———	....	....
510	.....	.....	.....	.....	8.03	13.47	.....	.....	.....	.....	.....	.....	———	....	....
511	.....	.....	.....	.....	.....	.....	7.83	14.37	.....	.....	.....	.....	———	....	....
301	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	5.62	7.45	Corrected for superheat	10 pipes...	72.9
302	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	5.78	9.92	———	....	....
303	.....	.....	.....	.....	.....	.....	.....	.....	6.31	8.45	.....	.....	....	....	57.4
605	.....	.....	.....	.....	8.02	15.53	.....	.....	.....	.....	.....	.....	———	....	....
512	.....	.....	.....	.....	.....	.....	7.71	24.00	.....	.....	.....	.....	———	....	....
606	.....	.....	.....	.....	.....	.....	.....	.....	6.18	16.10	.....	.....	———	....	....
513	.....	.....	.....	.....	.....	.....	.....	.....	6.45	11.32	.....	.....	———	....	....
515	.....	.....	.....	.....	.....	.....	.....	.....	6.83	13.82	.....	.....	———	....	....
516	.....	.....	.....	.....	.....	.....	.....	.....	6.94	13.60	.....	.....	———	....	....

TABLE 8—Continued.

Comparison of Total Tonnage Obtained for Each Brine Cooler Section and Mean Temperature Difference between Brine and Liquid Ammonia in Cooler.

No. of Test.	TONNAGE DIVISION.												Tests Discarded on Account of Excessive t_d Showing Superheating of Vapor in Top Pipes for Which Corrections Could Not be Made.	Number of Pipes that Were Found to be Effective from Temp. Readings in Ammonia Return Bends at Rear of Coolers.	k. Heat Transfer Coefficients Used for Plotting Acceptable Points on Plate 9.
	13 to 14		9 to 10		8 to 9		7 to 8		6 to 7		5 to 6				
	Actual Tonnage.	t_d	Actual Tonnage.	t_d	Actual Tonnage.	t_d	Actual Tonnage.	t_d	Actual Tonnage.	t_d	Actual Tonnage.	t_d			
2	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56
408	.....	.....	.....	.....	.....	.....	.....	.....	6.35	11.28	.....	.....	—	....	....
403	.....	.....	9.42	12.67	.....	.....	.....	.....	.....	.....	.....	.....	....	....	59.9
409	.....	.....	.....	.....	.....	.....	7.41	12.20	.....	.....	.....	.....	—	....	....
404	.....	.....	.....	.....	.....	.....	.....	.....	6.17	9.73	.....	.....	....	....	51.0
406	.....	.....	.....	.....	.....	.....	7.13	9.41	.....	.....	.....	.....	....	....	60.9
304	.....	.....	.....	.....	.....	.....	.....	.....	6.31	9.69	.....	.....	....	....	52.4
407	.....	.....	.....	.....	.....	.....	7.90	10.17	.....	.....	.....	.....	....	....	62.5
405	.....	.....	.....	.....	.....	.....	7.68	9.92	.....	.....	.....	.....	....	....	62.2
410	.....	.....	.....	.....	.....	.....	7.95	10.34	.....	.....	.....	.....	....	....	61.8
305	.....	.....	.....	.....	.....	.....	7.72	9.39	.....	.....	.....	.....	....	....	66.0
607	13.29	25.00	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	—	....	....

value of k would be high. Taking the lowest value of $t_b = 5.8$, found for test No. 502, the corresponding value of $k = 73.4$, column 25, Table 4. For the lowest value of $k = 25.8$, test No. 512, the corresponding value of $t_d = 24.00$. The temperatures of the vapor entering the accumulator, or, which is the same, the temperature of the vapor leaving the brine coolers, showed the reason for this great variation in the value of t_d. Readings taken of the temperature of the ammonia flowing through the rear end return bends made it possible to make a few corrections in the value of k as indicated in Table 8.

Then by dividing the acceptable tests, for which the values of k are given in column 56, Table 8, into two groups, for tonnages varying between 5 and 6, also between 6 and 8, and plotting them against their respective relative mean speeds of ammonia vapor and brine, column 39, Table 7, the two curves shown on Plate 9 could be assumed to represent in a general way the variation of the rate of heat transfer with a change in the speed of the ammonia vapor and brine in a double pipe brine cooler of the type tested. The division in two groups was made, as the test date showed conclusively that for the position of the accumulator used in reference to the brine coolers best ressults were obtained at from 5 to 6 tons refrigeration for each cooler section. The equation for Curve A was found to be:

$$k = 20.6 \sqrt{0.521\ v_a + v_b}, \qquad \text{(Equation 10)}$$

and for Curve B:

$$k = 15.2 \sqrt{0.521\ v_a + v_b}. \qquad \text{(Equation 11)}$$

Equation 10 applies to such cases where the liquid feeding system is so arranged that saturated ammonia vapor is discharged from the brine coolers at all times where the coolers are operated at the maximum capacity for which they were installed and any lower tonnage.

CONCLUSIONS

To obtain results over greater ranges of tonnage and consequent greater range of ammonia vapor speed, a test plant should be equipped either with hand-operated expansion valves for each cooler stand or else an accumulator the height of which in reference to the brine coolers can be varied so that saturated vapor is discharged from the coolers. Thermometers placed in mercury wells located in each ammonia return bend should be observed continuously to assure the presence of some liquid ammonia in each cooler pipe to the discharge end.

Care should be had in so placing the brine inlet and outlet thermometers to assure thorough mixing of the brine as it comes from the brine heaters and various brine cooler sections, so as to obtain the true brine range.

The plant should also be equipped with mercury columns so that the ammonia pressures at the inlet and outlet of the coolers can be accurately obtained. Excessive height of accumulator or excessive height of cooler to obtain large capacity and high vapor speeds often lead to results opposite to those desired on account of the great ammonia pressure difference between the bottom and the top of the cooler raising the mean temperature of the boiling ammonia in proportion more than the increased vapor speed helps to increase the value of k. High brine temperature ranges can, for this reason, only be secured by using several banks of coolers through which the brine passes in series and the ammonia in parallel.

By comparing equation 10 with similar formulæ deduced for ammonia condensers in a paper read at the December, 1914, meeting of The American Society of Refrigerating Engineers, and published in Vol. 1, No. 1, of the A. S. R. E. JOURNAL, it will be found, taking into consideration that the latter formulæ are based on inlet ammonia vapor speeds, that the constants for liquefaction and evaporation of ammonia are practically the same and will prove to be the same as soon as a way is found to make more accurate tests and with it more conclusive analyses.

We are indebted to Mr. J. Harold McCreery for his valuable assistance in completing the calculation of the test results and compiling the tables.

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SOCIETY AFFAIRS

PROGRAM

Joint Meeting

of

The American Society of Refrigerating Engineers

and

American Association of Refrigeration

The Fourth Western Meeting of The American Society of Refrigerating Engineers and a special meeting of the American Association of Refrigeration will be held jointly in San Francisco, Cal., on September 21 to September 25, 1915.

In the last three issues of the A. S. R. E. JOURNAL is the itinerary of a special train movement on account of this meeting. The special train will leave Chicago, Ill., on Wednesday, September 15, at 6:10 o'clock P. M., and will arrive in San Francisco, Cal., on Tuesday evening, September 21, at 5:45 o'clock P. M.

The headquarters of both associations will be at the Inside Inn, within the grounds of the Panama-Pacific International Exposition. The rates for rooms, European plan, include unlimited admission to the Exposition grounds during stay at this hotel. All rooms are outside rooms, with sunlight during some portion of the day, and the hotel accommodations are first class in every respect. The rates are as follows:

Rooms without bath, one person in room, $2.00 per day; two or three persons in one room, $1.50 per person per day.

Rooms with bath, one person in room, $3.00 to $4.00 per day, according to location; two persons in one room, $2.00 to $2.50 per person per day, according to location.

The sessions of our meeting, as well as the International Engineering Congress, will be held in the Exposition Memorial Auditorium at the Civic Center in San Francisco.

The entertainment program and list of papers to be read and topics discussed are as follows:

TUESDAY, SEPTEMBER 21

Evening, 8 o'clock—Dinner at Portola-Louvre Café, Market and Powell streets, $2.50 per plate.

WEDNESDAY, SEPTEMBER 22

Morning, 9 o'clock—Sightseeing trip around San Francisco, viewing Golden Gate Park, Cliff House and Presidio, and returning by the Exposition grounds. Price, $1.00 per person. Time, 2½ hours. Luncheon at the Portola-Louvre Café, 50 cents per person. Board cars at Inside Inn.

Afternoon, 1 o'clock—Sightseeing trip from San Francisco via Creek route ferry to Oakland, Berkeley, Claremont, Piedmont and Lake Merritt to Canyon Inn, Hayward, and returning via Foothill Boulevard. Price, including dinner at Canyon Inn, $3.50 per person. Board cars at Portola-Louvre Café.

THURSDAY, SEPTEMBER 23

Morning, 9:30 o'clock—Joint meeting of A.S.R.E. and A.A.R.

Roll call.

Reading of minutes of Tenth Annual Meeting of A.S.R.E.

Report of Tellers of Election of Members of A.S.R.E.

Address by Frank A. Horne, New York, N. Y., president of A.A.R.

Paper on "Oil Engine Driven Raw Water Ice Manufacturing Plant." Louis K. Doelling, New York, N. Y., president of A.S.R.E.

Discussion of Topic—Raw Water Ice Making Systems.

Afternoon, 2 o'clock—Attend session of International Engineering Congress in Exposition Memorial Auditorium, when the following papers on refrigeration will be presented:

Development of Refrigeration in France. L. Marchis, Paris, France.

Development of Refrigeration in Sweden. Thor Andersson, Stockholm, Sweden.

Development of Refrigeration in the United States. J. F. Nickerson, Chicago, Ill.

FRIDAY, SEPTEMBER 24

Morning, 9:30 o'clock—Joint meeting of A. S. R. E. and A. A. R.

Paper on "Fruit Precooling and Precooling Plants." W. C. Phillips, San Francisco, Cal.

Paper on "The San Bernardino Precooling Plant." C. M. Gay, Los Angeles, Cal.

Paper on "Rate of Heat Transfer in Double Pipe Brine Coolers." Van R. H. Greene and Fred Ophüls, New York, N. Y.

Discussion of Topic—Power Required to Agitate Brine in Can Ice Tanks.

Afternoon—Refrigeration day at Exposition.

Evening, 7 o'clock—Dinner at Inside Inn, $2.00 per person.

SATURDAY, SEPTEMBER 25

Afternoon, 1:30 o'clock—Steamboat excursion around the Bay of San Francisco, returning at 6 o'clock. All principal points of interest will be visited, including a trip through the Golden Gate. Price, $1.00 per person, providing a party of one hundred can be made up. Board steamer at dock of Crowley Launch Company on the Escadero near ferry depot.

Evening, 8 o'clock—Banquet at the Hof Brau Café, Fourth and Market streets, $5.00 per cover, including wine.

COUNCIL NOTES

The Council held a meeting in the office of the Society, 154 Nassau Street, New York City, on Thursday afternoon, August 12, 1915, with the following officers and directors in attendance: Messrs. Doelling, Shipley, Greene, Behn, Nickerson, Horne, Frick, Torrance, Matthews and Ross.

The minutes of the meeting of The Council of January 29, 1915, were read and approved.

The Membership Committee presented applications for membership from the following, and they were approved in the grades indicated:

MEMBERS

Charles W. Berry, Boston, Mass.
Frederic T. Brandt, Waynesboro, Pa.
Charles I. Day, Jacksonville, Fla.
Eric H. Peterson, Detroit, Mich.
George B. Stacy, San Francisco, Cal.
Edgar Steiner, St. Louis, Mo.
Ralph W. Bowers, Waynesboro, Pa.

ASSOCIATE MEMBERS

C. Thomas Baker, Jacksonville, Fla.
Alan K. Gillespie, St. Louis, Mo.
S. T. McAdam, Pensacola, Fla.
J. Malcolm May, Wellington, N. Z.
George W. Mitchell, St. Louis, Mo.

JUNIOR MEMBERS

Harry C. Bruce, St. Louis, Mo.
Karl R. Dieterle, Los Angles, Cal.
Frank B. Higgins, Canton, Ohio.
Stewart E. Lauer, York, Pa.
Henry R. Rosebro, Jacksonville, Fla.
Guy V. Thompson, Portsmouth, Ohio.
Carl J. Weaver, Canton, Ohio.

It was the sense of The Council that a superintending engineer in charge of several plants is eligible to membership in the grade of Associate Member only, unless he possess special technical knowledge.

George H. Geisler, Oakland, Cal., was elected a Director until the annual meeting of 1915, vice A. P. Criswell, deceased.

A complete set of the Transactions of the Society was ordered sent to the Franklin Institute, Philadelphia, Pa., and copies of the A. S. R. E. JOURNAL were ordered sent to the following publications, organizations and institutions in exchange for publications issued by them:

American Society of Civil Engineers, New York, N. Y .
Franklin Institute, Philadelphia, Pa.
University of Pennsylvania, Philadelphia, Pa.
Columbia University, New York, N. Y.
St. Louis Public Library, St. Louis, Mo.
American Society of Heating and Ventilating Engineers, New York, N. Y.
Bulletin of the Russian Refrigeration Committee, Petrograd, Russia.
Zeitschrift fur Eis und Kaelte-Industrie, Vienna, Austria.
Zeitschrift fur die gesamte Kaelte-Industrie, Munich, Germany.

President Doelling reported the death of the following members since the last annual meeting:

Benjamin F. Daly, Rockaway Beach, N. Y.
A. P. Criswell, Chicago, Ill.
James Edwin Quigley, Pittsburgh, Pa.
William J. Webber, Boston, Mass.

The matter of the Society's cooperation with the Joint Committee on Classification of Technical Literature was referred to the next meeting of The Council.

The Secretary was authorized to purchase folders containing bird's-eye view of the Panama-Pacific International Exposition and a map of San Francisco, on which the program of our Fourth Western Meeting is to be printed, at a cost not exceeding $17.

The Secretary was authorized to purchase for the library a complete file of *Zeitschrift fur Eis und Kaelte-Industrie,* Vienna, Austria, as well as a complete file of *Zeitschrift fur die gesamte Kaelte-Industrie,* Munich, Germany.

President Doelling reported that the Society had been invited to send a delegate to the Second Pan-American Schientific Congress, to be held in Washington, D. C., on December 27, 1915, to January 8, 1916, and further reported that he had appointed N. H. Hiller as the delegate of the Society.

The Society's membership dues of $10 to the American Association of Refrigeration were approved and ordered paid.

The following resignations were accepted with regret:

George Berna, Ithaca, N. Y.
H. M. Moore, Des Moines, Ia.
Samuel R. Frantz, Waynesboro, Pa.
Jaroslav Spinka, St. Louis, Mo.
J. C. Bertsch, New York, N. Y.

W. H. Ross, *Secretary.*

AMENDMENTS PROPOSED TO THE CONSTITUTION

At the last annual meeting the President, Henry Torrance, on recommendation of The Council, proposed the following amendments to paragraphs C 17 and C 20 of the Constitution, subject to discussion and such modification as may be accepted by the proposer:

(Words in italics indicate new matter; words in parentheses to be omitted.)

INITIATION FEES AND DUES

C 17. The initiation fee for membership shall be as follows:

For Members and Associates, $5.

For Juniors, $5. A Junior on promotion to any other grade of membership shall pay an additional fee of $5.

The annual dues for membership shall be as follows:

For Members and Associates *residing in the United States and Canada,* $15; *elsewhere, $10.*

For Juniors *residing in the United States and Canada,* $10; *elsewhere, $5* (for the first six years of their membership, and thereafter the same as for a Member or Associate).

THE COUNCIL

C 20. The affairs of the Society shall be managed by a Board of Directors chosen from among its Members and Associates, which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, the two Vice-Presidents, Treasurer, (and) fifteen Members and Associates, *and the last five living Ex-Presidents, ex-officio.* Seven members of The Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of The Council, but shall not have a vote therein.

These proposed amendments will come before the Eleventh Annual Meeting for final discussion and action.

JOINT MEETING OF A. A. R. AND A. S. R. E.

Since the July number of the A. S. R. E. Journal was issued arrangements have been made with the American Association of Refrigeration for a joint meeting in San Francisco on the dates of our Fourth Western Meeting on

September 21 and 25. A joint program has been prepared, which appears on another page of this number of The Journal.

On the afternoon of September 23 members and guests of both associations will attend a session of the International Engineering Congress, when its papers on refrigeration will be presented.

The complete program affords the members of both associations an opportunity to hear and discuss several exceptionally interesting papers, as well as to participate in several very delightful excursions. President Doelling is anxious that as many members of the Society attend this meeting as possible.

The headquarters of both associations will be at the Inside Inn, within the grounds of the Panama-Pacific International Exposition.

A. S. M. E. BOILER CODE

Arrangements have been made with the American Society of Mechanical Engineers whereby members of The American Society of Refrigerating Engineers may secure copies of the A. S. M. E. Boiler Code on the construction of stationary boilers and allowable pressures at 80 cents per copy.

This boiler code was reviewed in the March issue of the A. S. R. E. JOURNAL. Since the code was published it has been adopted by a number of States and municipalities as an official code.

Nearly every member of the Society is interested in the operation, construction and design of boilers, and it is urged that they secure a copy of this publication.

Orders may be sent direct to the American Society of Mechanical Engineers, 29 West 39th Street, New York City, or may be ordered through our Secretary.

TRADE CATALOGS

(Below is given a list of trade catalogs, etc., received at the Society's office since the publication of the last issue of the A. S. R. E. Journal.)

Estate of W. M. Schwenker, New York, N. Y.—The Protecto Ammonia Helmet.

Stevenson Cold Storage Door Company, Chester, Pa.—The Door which Keeps the Cold in Cold Storage.

Builders Iron Foundry, Providence, R. I.—Venturi Hot Water Meter, Bulletin No. 85.

Builders Iron Foundry, Providence, R. I.—The Venturi Meter for Pump Discharge Lines, Refrigerating Plants, etc., Bulletin No. 84.

National Tube Company, Pittsburgh, Pa.—Index for "National" Bulletins Nos. 1 to 20.

McCord Manufacturing Company, Detroit, Mich.—McCord Force Feed Lubricator.

WHAT OTHERS SAY

"The great American Society of Refrigerating Engineers is now publishing a bi-monthly journal containing its transactions. The first number has been received and indicates an essentially scientific publication.

* * * * * *

"It must be admitted that the new periodical fills a great gap in American literature on refrigeration, because it is the first to publish extended articles of a scientific character, while the trade journals confine themselves to material more easily understood. It is very important for the development of every industry to have an organ in which scientific thought may be expressed and criticised, because only by this means is it possible to instill scientific reasoning into actual practice. The necessity for such research will readily be admitted by all who are broad minded enough to see beyond the present scope of their own business."—*Zeitschrift fur die gesamte Kaelte-Industrie,* Munich, Germany.

NOTICE

The Society subscribes for the Monthly Catalogue of United States Public Documents, which is the only publication giving a complete list of all the Government's publications as they appear. Since THE JOURNAL can find space to review only a few of those of the many valuable monographs and publications issued by the various departments that will be found of most general interest to its readers, it is to be hoped that members seeking information along certain lines will make use of this catalogue, files of which will be found in the Society's office. On request, the Secretary will undertake to advise a member what publications have appeared on a given subject during the current year, and at what price they can be obtained from the Superintendent of Documents, Washington, D. C.

A CORRECTION

On pages 46 and 47 of the July number of the A. S. R. E. JOURNAL there appeared a table on "Cost of Manufacturing Raw Water Ice," taken from the *General Electric Review* of July, 1914. We neglected to state that this paper was prepared by the Arctic Ice Machine Company, Canton, Ohio, and take this means of giving them proper credit for the table.

WANTED

Anyone knowing the present address of any of the following members is requested to communicate with the Secretary:

E. L. FLORY OSCAR DREYER D. L. HOLDEN

A. S. R. E. Journal

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PUBLISHED BI-MONTHLY

BY

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

154 NASSAU STREET, NEW YORK, N. Y.

Subscription price, United States and dependencies, $3.00 a year; Foreign countries in Postal Union, $4.00 a year.

Application for entry as second-class matter at the Post Office at New York, N. Y., pending.

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A. S. R. E. Journal

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FLOW OF SUPERHEATED AMMONIA THROUGH ORIFICES*

CALCULATION OF SIZES OF AMMONIA SAFETY VALVES

BY EDWARD F. MILLER, BOSTON, MASS.

This article gives in brief the results of experimental work made at the instigation of the Massachusetts State Board of Boiler Rules and carried on by F. L. Fairbanks, L. Williams and E. F. Miller, members of the Committee on Municipal and State Regulations for Refrigerating Plants and Refrigerants, appointed by The American Society of Refrigerating Engineers, to keep in touch with such work. Professor Carlton A. Read, of Worcester Polytechnic Institute, and G. H. Clark, of the Massachusetts Institute of Technology, took an active part in the work and assisted both in the experimental work and in the calculations

Your committee, at one of its early meetings, decided that its first step should be toward securing data on the flow of superheated ammonia through orifices.

The Quincy Market Storage & Warehouse Company, through Mr. Fairbanks, kindly offered us for such tests the use of two machines at their Eastern avenue plant. The committee accepted their offer, and hereby takes occasion to express its appreciation of the numerous favors extended it by the Warehouse Company and by Mr. Fairbanks.

Orifices of different diameters, of different lengths and with different radii of entrance curves were tested, as reported in the summary, which is appended.

The gas discharge through the orifice was taken into the suction side of a 1,000-ton machine, which was run just to take care of this gas.

This paper is copyrighted by The American Society of Refrigerating Engineers and issued to the membership prior to the meeting on December 6, 7 and 8, 1915, at which it will be read, in order that members may prepare any discussion which they may wish to present. It is issued in confidence and with the understanding that the paper is not to be reprinted, even in abstract, until after it has been read at the meeting, and then only on the authority of the Publication Committee. It is subject to revision by the author and the Society.

The Society is not responsible for statements or opinions advanced in papers or in discussions at its meetings.

The gas entering the orifice was discharged from a 26 x 48-inch twin cylinder, single-acting compressor, which was run at speeds varying from 20 to 54 revolutions per minute.

The temperature and pressure of the gas entering the compressor were noted.

The weight of ammonia gas flowing through the orifice per minute was calculated (after the pressure in the orifice had been determined) by making use of the theoretical formula given later. Then the volume of this weight, reduced to suction pressure and temperature, was figured and the volumetric efficiency of the compressor determined. This volumetric efficiency was calculated merely as a matter of interest, although it did serve as a rough method of checking the results.

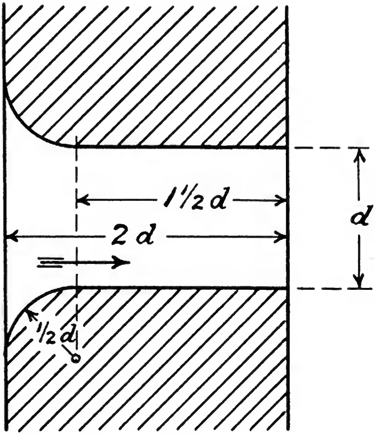

Fig. 1

It appeared that with an orifice made as per Fig. 1 the absolute pressure in the orifice was 0.585 that of the absolute pressure at entrance.

For such orifices this simple empirical formula—

$$\text{Weight per sec.} = \frac{0.4 \times \text{area orifice in sq. ins.} \times \text{head press. in lbs. sq. in. abs.}}{\sqrt{460 + \text{head temp. in degrees F.}}}$$

was found to give reasonable accuracy, and was made use of in calculating the discharge capacity of safety valves.

DISCHARGE OF AMMONIA SAFETY VALVES

$$\frac{C \times \pi \times d \times d \times 0.707 \times 290 \times 60 \times 0.4}{6\sqrt{700}} = \frac{\text{piston displ. cu. ft. min.}}{5.665}$$

$$\text{Piston displ. cu. ft. min.} = \frac{C \times d^2 \times \pi \times 0.707 \times 290 \times 60 \times 0.4 \times 5.665}{6\sqrt{700}}$$

$$\text{Piston displ. cu. ft. min.} = 551.67 \times C \times d^2.$$

C is a coefficient used to allow for shape of orifice. It was taken as 0.85 on a ½" valve, 0.90 on a ¾" valve, 0.93 on a 1" valve, and 0.96 on sizes of 1¼" and over.

290 is the head pressure (absolute).

5.665 is the volume of one pound of saturated ammonia at 50 pounds absolute, which was considered as sufficiently high for suction pressure. The fact that the displacement efficiency of the compressor is less than 100 per cent gives an additional margin of safety to the valve. The factor 0.707 is the sine of a 45-degree angle, the effective lift being only this proportion of the actual lift on a 45-degree bevel-seated valve. The vertical lift of the valve is taken as one-sixth of the diameter; 700 is taken as the absolute temperature of the ammonia gas leaving the compressor, or 460 plus 240°, the reading as shown in degrees F.

TABLE I.—RELIEVING CAPACITY OF AMMONIA SAFETY VALVES AT 290 LBS. ABSOLUTE PRESSURE EXPRESSED IN TERMS OF THE PISTON DISPLACEMENT OF COMPRESSOR IN CUBIC FEET PER MINUTE.

Diameter of Valve.	Cubic Feet Piston Displ. per Min.
½ inch	117.
¾ "	279.
1 "	513.
1¼ "	828.
1½ "	1192.
2 "	2118.

These values are shown on diagram, Fig. 2.

The diameter of the discharge outlet and of a discharge pipe 100 feet in length, which would be required to take care of this gas with 5 pounds accumulation in pressure at the entrance end and atmospheric at exit end was figured by the Babcock formula, and the results are shown by the plot, Fig. 3.

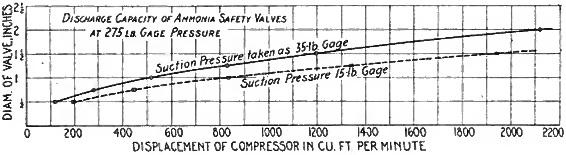

Fig. 2

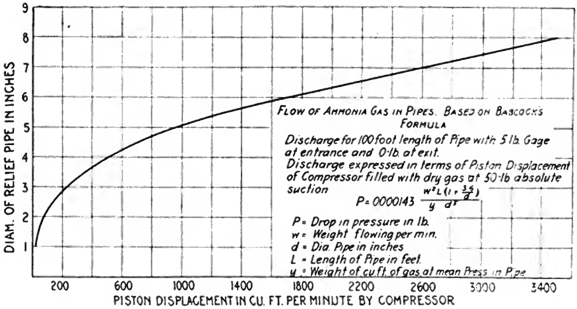

Fig. 3

It at once became evident that it was impossible to use discharge pipes of the large diameters called for by this plot.

The committee then decided to make a safety valve discharging into a pipe where the pressure might accumulate at time of maximum discharge to 0.585 the entrance pressure, or with 275 pounds gage entrance pressure to 154 pounds gage.

As the weight passing through a valve, or through an orifice, is just the same in amount with 154 pounds as back pressure as with atmospheric pressure as back pressure, the entrance pressure in each case being 275 pounds gage, the size of the valve need not be increased and the discharge pipe conveying the gases through the roof may be made much smaller.

In such a valve the back pressure must be kept from acting on the back of the valve. Such a valve has been made by Mr. Fairbanks and was tested out on his machine.

The same idea has been embodied in a new steam safety valve, which on test was found to work perfectly.

Your committee found available no data from which to calculate the diameter of a pipe of given length, which is to discharge a given weight of gas per minute entering at 154 pounds gage and leaving at atmospheric pressure. Therefore, a run of piping was erected in the engine room at the Eastern avenue plant and some data obtained. A straight run of 2" pipe 260 feet long was also installed at the Massachusetts Institute of Technology, and a known weight of saturated steam at high pressure sent into one end of the pipe, the other end being open.

Sufficient data was obtained from these different experiments to warrant the committee in fixing the sizes of the discharge pipes as shown in Table II.

TABLE II.—SIZES OF DISCHARGE PIPES.

Piston Displacement Cubic Feet Per Min.	Diameter of Valve.	Diameter of Discharge Pipe for First 100 Feet.
120	½ inch	1
280	¾ "	1¼
510	1 "	1½
830	1¼ "	2
1200	1½ "	2½
2120	2 "	3

For the second 100 feet the pipe should increase one pipe size, and so on for each additional 100 feet. Pipe bent to a radius at least as great as five pipe diameters is to be used when it is necessary to turn a corner.

A number of commercial ammonia safety valves were tested by the committee. Some of these were unreliable in their action and unsatisfactory in many ways.

BRIEF SUMMARY OF TESTS ON FLOW OF SUPERHEATED AMMONIA GAS THROUGH AN ORIFICE

On February 26th, 1915, experiments were made at head presures varying from 200 to 218 pounds absolute, 100 degrees superheat on an orifice 1.0045″ diameter in a plate ¾″ thick. The entrance to the orifice was rounded with a smooth curve of ¼″ radius. The pressure in the orifice was taken through a hole 1/32″ diameter drilled at right angles to the surface of the straight part of the orifice at a distance of ¼″ from the exit edge.

The ratio of orifice pressure to head pressure (both absolute) was 0.362 to 1. This same orifice was then drilled so that the pressure could be measured 15/32″ from the entrance face. There the ratio of orifice pressure to head pressure was 0.40 to 1.

On March 11, 1915, an orifice 1″ diameter in a plate 1½″ thick was tried. This orifice was made with a smooth curve, at entrance, of ½″ radius. The pressure in the orifice was taken through a hole 1/32″ diameter located 33/64″ from the entrance face. The straight part of the orifice was 1″ long. Pressures at entrance were varied from 84 pounds absolute to 179 pounds absolute, and the superheat from 118 to 160 degrees. The ratio of orifice to head pressure, taking the average of all the readings, was 0.572 to 1. If the two last readings were omitted (these evidently being in error) the average ratio becomes 0.575. Table III contains the results of these tests.

TABLE IV. FLOW OF SUPERHEATED AMMONIA.
Entrance radius of orifice, ¼ inch. Tests of March 16, 1915.

Head Pressure Pounds Absolute.	Head Temperature, °F.	Orifice Pressure Pounds Absolute.	Ratio Orifice Press. to Head Pressure.
152	223	72.6	0.478
129	221	61.6	0.476
110	216	52.1	0.473

TABLE III. FLOW OF SUPERHEATED AMMONIA THROUGH AN ORIFICE, 1 INCH DIAM., 1½ INCHES LONG.

Tests of March 11, 1915. Compressor used, 26 x 48 inches, twin cylinder, single acting.

Head Press. Lbs. Abs.	Head Temp. °F.	Temp. Sat. Ammonia at Head Pressure. °F.	Degrees of Superheat. °F.	Orifice Pressure Lbs. Abs.	Back Pressure Lbs. Abs.	Temp. beyond Orifice °F.	Revs. of Compressor per Min.	Ratio Head Pressure to Orifice Pressure.	Heat Contents of Vapor Entering Orifice. B. t. u. per Lb.	Heat Contents of Vapor in Orifice. B. t. u. per Lb.	Velocity in orifice in Feet Per Second.	Volume of 1 Lb. at Orifice Condition.	Lbs. Discharged Per Second.	Volum of 1 Lb., Entering Orifice.
153.7	200	79.9	120.1	87.0	24.5	178	46	0.572	634	595	1400	3.90	1.956	2.56
156.7	210	81.1	118.9	89.0	24.5	187	46	0.573	634	595	1400	3.85	1.980	2.50
160.7	220	82.6	137.4	92.0	24.5	195	46	0.573	645	604	1418	3.78	2.040	2.52
162.7	226	83.3	132.7	94.0	24.5	202	46	0.577	...	...	....	....	....	....
160.7	230	82.6	147.4	93.0	24.5	206	46	0.578	...	...	....	....	....	....
160.7	228	82.6	145.4	92.5	24.5	204	46	0.575	...	...	....	....	....	....
178.7	235	88.9	146.1	104.0	26.5	209	54	0.582	654	615	1400	3.53	2.162	2.30
177.7	240	88.6	151.4	103.0	26.5	216	54	0.580	...	...	....	....	....	....
175.7	246	87.9	158.1	101.0	26.5	221	54	0.575	...	...	....	....	....	....
177.7	248	88.6	159.4	102.5	26.5	224	54	0.575	...	...	....	....	....	....
115.7	228	63.9	154.1	66.0	18.5	210	32	0.570	...	...	....	....	....	....
116.7	220	64.4	155.6	67.0	19.5	204	32	0.574	...	...	....	....	....	....
116.7	210	64.4	145.6	67.0	19.5	194	31	0.574	642	602	1414	5.39	1.430	3.51
100.7	193	56.4	136.6	57.5	18.5	180	27	0.571	...	...	....	....	....	....
100.2	188	56.1	131.9	57.0	17.5	174	26.5	0.569	...	...	....	....	....	....
81.2	171	45.2	125.8	44.5	16.5	160	20.5	0.548	...	...	....	....	....	....
83.7	165	46.8	118.2	46.5	16.5	153	20.5	0.556	...	...	....	....	....	....

On March 16, 1915, experiments were made on an orifice 1″ diameter, 2″ face to face of flange, with curve ¼″ radius at entrance to orifice, and with orifice pressure measured 9/32″ from entrance face. As shown by Table IV, this orifice gave a ratio of orifice pressure to head pressure averaging 0.476 to 1.

The orifice was then changed so as to have the curve at entrance ¾″ radius, leaving the straight part of the bore 1¼″ long. The pressure in the orifice was taken through a hole 1/32″ diameter, drilled at right angles to the bore and 25/32″ in from the entrance face. The observations with the orifice of larger entrance radius are given in Table V. The ratio of orifice pressure to head pressure was then found to average 0.585.

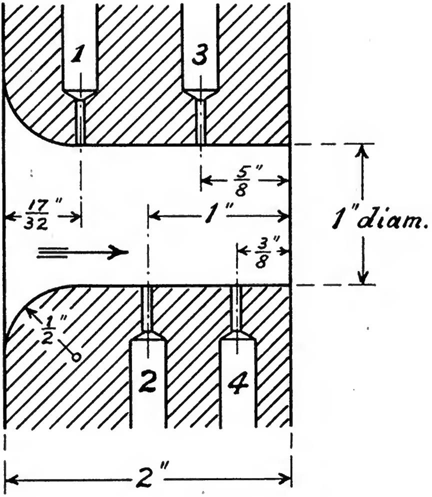

Fig. 4

These tests were made with head pressures varying from 84 to 160 pounds absolute and with from 85 to 116 degrees superheat.

On this same date another orifice was tried. This orifice, Fig. 4, was 2″, face to face of flange, with 1″ diameter hole, having entrance curve ½″ radius. Pressures in the orifice were taken at four places 17/32″ back from the front face, and ⅜″, ⅝″ and 1″ in from the back face. The pressures read from the opening ⅜″ in from the back face were from one to two pounds lower than the readings

TABLE V. FLOW OF SUPERHEATED AMMONIA. ENTRANCE RADIUS OF ORIFICE, ¾ INCH.
Tests of March 16, 1915.

Head Pressure Lbs. Absolute.	Head Temp. °F.	Head Saturation Temp. °F.	Degrees of Superheat. °F.	Orifice Pressure Pounds Absolute.	Ratio of Orifice Pressure to Head Pressure.	Temp. Beyond Orifice. °F.	Suction Pressure Pounds Absolute in Pipe near Compressor.	Suction Temp. in Pipe near Compressor. °F.	R.P.M. of Compressor.	Displ. of Compressor Cu. Ft. Min.	Pounds Ammonia Discharged Per Second by Orifice.	Vol. Per Lb. at Suction Pressure and Temperature.	Vol. Discharged by Orifice per Min. at Suction Press and Temp.	Ratio of Vol. Discharged to Compressor Displ.
87	155	48.8	106.2	50.6	0.582	138	30.6	0.0	24	708	1.48	...	...	...
84	150	47.0	103.0	49.6	0.590	136	30.6	1.5	24	708	1.07	...	...	....
85	147	47.6	99.4	50.1	0.590	133	30.6	1.5	24	708	1.10	...	...	....
118	155	64.9	90.1	70.1	0.594	132	30.1	—0.5	34	1053	1.52	...	...	....
118	158	64.9	93.1	69.6	0.580	136	30.1	—0.5	34	1053	1.53	...	...	....
160	168	82.3	85.7	92.6	0.578	142	29.3	—1.3	46	1358	2.04	9.5	1163	0.857
158	176	81.5	94.5	92.1	0.583	147	29.1	—1.3	46	1358	2.00	9.5	1140	0.840
156	197	80.8	116.2	91.1	0.583	172	28.6	4.3	46	1358	1.95	9.8	1146	0.842

TABLE VI. FLOW OF SUPERHEATED AMMONIA. ENTRANCE RADIUS OF ORIFICE, ½ INCH. (FIG. 4.)
Tests of March 16, 1915. Compressor used, 26 x 48 inches, twin cylinder, single acting.

Head ess. Lbs. Abs.	Head Temp. °F.	Head Saturation Temp. °F.	Degrees of Superheat. °F.	Orifice Pressure, Pounds Absolute				Ratio Orifice Press. Pt. (1) to Head Press.	Back Pressure Lbs. Absolute.	Temp. Beyond Orifice. °F.	Suction Press. in Pipe Near Compressor. Lbs. Absolute	Suction Temp. °F.	R.P.M. of Compressor.	Displ. Compres. Cu. Ft. Min.	Vel. Through Orifice. Feet per sec.	Pounds Ammonia Disch. per Second	Vol. per Lb. Am. at Suct. Press and Temp.	Vol. Am. Disch. Through Orifice per Min. at Suct. Press. and Temp.	Ratio of Vol. Disch. to Compressor Disp.
				Point (1)	Point (2)	Point (3)	Point (4)												
97	204	54.4	149.6	56.1	56.6	56.6	54.6	0.578	18.6	188	25.6	8.5	32	944	1418	1.22	11.1	813	0.862
98	209	55.0	154.0	55.6	56.6	56.6	55.6	0.578	18.6	195	25.6	8.5	32	944	1430	1.27	11.1	846	0.895
134	226	72.0	154.0	79.1	80.6	78.6	77.6	0.589	22.6	205	24.8	8.8	46	1358	1410	1.78	11.3	1228	0.905
133	234	71.6	162.4	78.1	78.6	77.6	76.8	0.600	22.6	217	24.6	8.8	46	1358	1418	1.77	11.4	1211	0.894
153	248	79.6	168.4	89.1	89.6	89.6	87.6	0.588	24.6	230	24.3	8.5	52	1534	1430	1.80	11.9	1285	0.835
152	253	79.3	173.7	88.1	88.6	89.6	86.6	0.578	25.6	233	24.3	8.3	54	1593	1437	1.84	11.9	1314	0.825

taken from the other places. The readings taken from the other openings agreed very closely. The ratio of throat pressure to head pressure, using the readings of throat pressure taken 17/32" from the front face, was 0.585 to 1. Table VI contains the results of these tests.

It would seem from these tests that the radius of the curve at entrance should be at least one-half the diameter of the orifice, and that the straight part of the orifice should be one and one-half diameters long.

With such an orifice the absolute pressure in the orifice may be assumed to be 0.585 of the absolute head pressure, provided the back pressure is not greater than 0.585 times the head pressure. If the back pressure is more than 0.585 times the head pressure, then the pressure in the orifice may be assumed to be a slight amount, perhaps one pound, above the back pressure.

The weight going through an orifice per second may be calculated thus:

V = velocity in feet per second.
a = area of orifice in square inches divided by 144, or the area of the orifice in square feet.
z = volume of one pound of ammonia gas at the throat pressure and condition in the orifice.
H_e = heat content of one pound of ammonia at head pressure and condition.
H_0 = heat content of one pound of ammonia at throat pressure and condition.

The condition in the orifice is found by assuming an adiabatic expansion from head pressure and condition to throat pressure.

$$V = 224 \sqrt{H_e - H_0}.$$

$$\frac{V \times a}{z} = \text{weight}.$$

To save interpolation in Goodenough & Mosher's tables,[1] two plots have been found useful. One plot, Fig. 5. gives the heat content H_e for any pressure and number of degrees of superheat. The other, Fig. 6, gives the volume for any pressure and superheat.

Knowing the head pressure and the degrees of superheat, the *Heat Contents* H_e can be read off from Fig. 5. The entropy is noted. Following down at constant entropy to the pressure in the orifice, the *Heat Contents* H_0 can be read; also the number of degrees the gas in the orifice is superheated.

On the second plot, Fig. 6, follow up on the same entropy to the pressure in the orifice and z, the volume of a pound at the condition in the orifice, can be read.

The simple empirical formula already mentioned, and giving results approximately correct for head pressures over 150 pounds

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[1] The Properties of Saturated and Superheated Ammonia Vapor. By G. A. Goodenough and William Earl Mosher. Bulletin No. 66, University of Illinois, Urbana, Ill., Jan. 27, 1913.

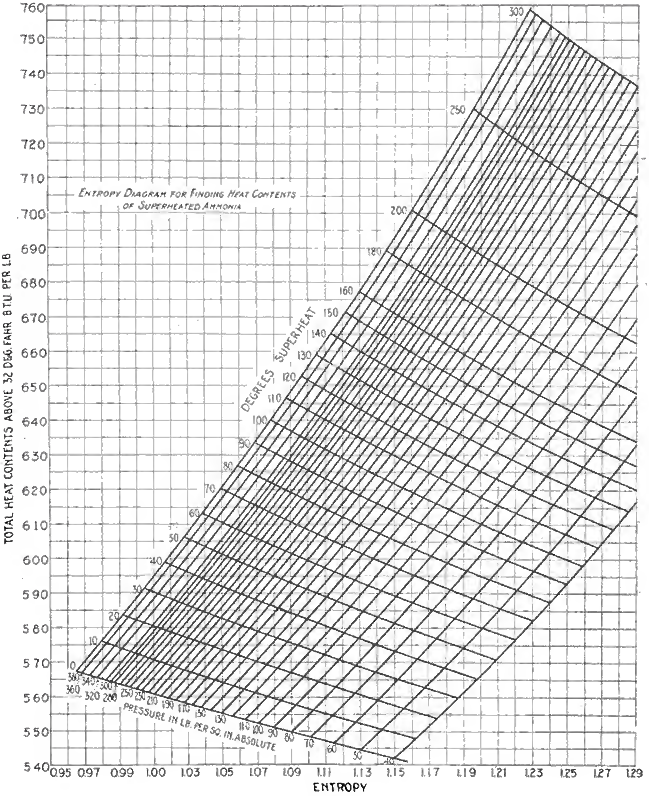
Entropy Diagram for Finding Heat Contents of Superheated Ammonia
TOTAL HEAT CONTENTS ABOVE 32 DEG. FAHR. B.T.U. PER LB
760
750
740
730
720
710
700
690
680
670
660
650
640
630
620
610
600
590
580
570
560
550
540
ENTROPY
0.95
0.97
0.99
1.00
1.03
1.05
1.07
1.09
1.11
1.13
1.15
1.17
1.19
1.21
1.23
1.25
1.27
1.29
DEGREES SUPERHEAT
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
180
200
250
300
PRESSURE IN LB. PER SQ. IN. ABSOLUTE
380
360
340
320
300
280
250
230
210
190
170
150
130
110
100
90
80
70
60
50
40

Fig. 5

noted Aug [illegible]

PRESENT STATUS OF BREWERY REFRIGERATION*

By Peter Neff, Canton, Ohio

It is apparent to all that but little attention at this time is being given by our engineers to the refrigeration of breweries.

The reason is obvious—there is not much new work being done. The industry is still a very large one, however, and refrigeration plays an important part in it.

Practically all existing breweries have refrigerating equipment in excess of actual requirements, and many of them operate at a higher cost than necessary.

The industry is in the position that, while it does not demand enlarged equipment, it is seeking to accomplish what is required with less operating expense and better conditions; here is our field of endeavor.

Lack of sufficient insulation is a common source of loss: Where rooms have to be rebuilt it is now usually practicable to install some modern type of insulation; even if no more than 3 inches of good insulation is used a very large saving will be effected. The insulation of exposed cold pipes will also help greatly. Where one machine can be made to do the work now done by two, the gains come fast.

I have in mind a brewery where two machines were constantly operated. During the month of June a shaft broke in one of these machines. The brewmaster was much worried, as it would require several weeks to install a new one. This occurrence gave them the cheapest lesson that they could have learned, for all that was necessary was a little more attention on the part of the engineer to successfully handle the work with one machine running full capacity, in place of the two machines running haphazard, as was formerly the case. We had tried to impress this fact upon them prior to the accident.

In many plants machines and other equipment have been added without giving proper attention to the connections. The loss of the liquid seal in receivers, with its consequent loss of plant efficiency, is as prevalent here as in other industries. Ancient customs and loose habits of operation are to be encountered. This can be changed only by the slow process of education.

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The subject of refrigeration for the attemperators must be treated in accordance with the individual ideas of the brewmaster, which differ. Some use a small attemperator requiring a low temperature and a rather high velocity of the circulating medium, while others accomplish the same result with higher temperature by the use of large attemperator coils. As a rule, you cannot change the custom, but you can alter the method employed to cool the circulating medium.

In a certain plant a large refrigerating machine had at times to be continuously operated to do a small amount of work in cooling brine for the attemperators. As this was being done in a small tank with practically no insulation, it may be readily imagined what the installation of a large well-insulated brine tank did for this plant.

Incidentally, exposed steam pipes and steam pumps are frequently to be found in refrigerated rooms. No one seems to be responsible for such things; they have always been there, and it therefore appears all right. There are also many opportunities for the improvement of the power equipment whereby savings may be effected.

By far the most interesting field for our work is in the fermenting room, where the problem of ventilation must be met. Gases produced by the fermentation must be gotten rid of. The usual method employed for this is to provide openings in the wall near the floor, through which a large amount of cold air continually passes, carrying the gases with it. An equivalent amount of outside air comes into the room at the same time, generally through an opening in the ceiling. In the summer this outside air is frequently ladened with moisture, which causes the ceiling to drip, mould to grow, and makes trouble for the brewer in many ways. Thousands of dollars have been expended on this problem with varying degrees of success. In one instance where the condition was exceptionally bad, a firm of brewery architects advised the raising of the ceiling some 8 or 10 feet, which involved an expenditure of approximately $10,000 without in any way overcoming the difficulty. An expenditure of less than $1,200 finally accomplished the desired result. Another method for removing the gases is to use an exhaust fan. This method increases the amount of moisture-ladened air introduced, with consequent disagreeable results.

A few years ago Mr. Starr mentioned the use of indirect refrigeration in the brewery, and referred to some work done by Mr. Livezey. Upon examination, I found this method a good one, and, with certain modifications, have employed it in several instances. The fact that temperatures below 32 degrees are not required makes possible the use of a bunker room without defrosting loss.

By effectually closing all openings in the fermenting room and, at the same time, providing for a continuous supply of fresh air, refrigeration and ventilation will be accomplished economically, and as a result the fermenting room will be so dry that painting can be carried on in it at any time.

Where other rooms are handled on the same circuit, as, for example, a shipping or racking room, sufficient fresh air can usually be obtained, while in the case where the fermenting room is to be handled alone openings with dampers may have to be installed.

Several things have to be considered in such installations, openings, such as stairways leading to floors below, are not desirable, as they permit the gases to settle in the lower rooms, for it is a well-known fact that such openings cannot be kept tightly closed. Then, too, the air circulation must not be such as to disturb the layer of gas on the top of the fermenting beer in open tubs. The results accomplished by this method, in addition to producing a sweet and dry fermenting room, are that less pipe surface will be required for the refrigeration of the room, owing to the rapid circulation of the air over the pipes; that these pipes may be kept free from a large accumulation of ice; that a saving in refrigeration losses may be effected by having regulated openings for ventilation, and that defrosting may be carried out without loss of efficiency. In fact, it frequently happens that for nights and Sundays sufficient refrigeration may be had from the ice accumulated on the pipes, and the necessity of operating the refrigerating machine continuously may be avoided. In one instance this system has been extended until it now practically takes care of the entire brewery.

It is impossible to lay down fixed rules for the amount of pipe, capacity of fan, size and arrangement of ducts, etc., as each installation offers different characteristics which must receive the careful attention of the engineer. In some instances the ordinary ventilating type of fan is all that is needed, while in others the larger type of blower must be installed. In figuring the ducts, the cost of installation and space available must be balanced against velocity and friction, entailing increased power consumption.

I find that brewery refrigeration is far from a dead issue. To-day offers a field for refrigerating engineering in which economies can be worked out that are a revelation to brewery owners.

Old prejudices and customs must be overcome with tact and perseverance which are just as important requisities as education and experience.

EFFECT OF VELOCITY AND HUMIDITY OF AIR ON HEAT TRANSMISSION THROUGH BUILDING MATERIALS*

By J. A. Moyer, State College, Pennsylvania

The work of the Engineering Experiment Station of the Pennsylvania State College in regard to the transmission of heat through various building materials has been devoted to a study of the effects produced by varying velocities and humidity of air over the material tested. Data of this kind have long been needed for the satisfactory and accurate design of heating and refrigerating systems.

All of the tests reported in this paper were made in the thermal testing plant of The Pennsylvania State College, where every effort has been made, without sparing expense, to eliminate all possible difficulties, so that accurate data may be obtained. The calorimeter is roomy, so that large samples can be tested. The means are also available for measuring and varying the velocity of the air; and by the use of accurate and delicate platinum resistance thermometers located in the calorimeter room, at the inside and outside walls of the material being tested, as well as on the surfaces, and imbedded at various distances in the body of the material, it is believed very accurate determinations can be made.

This plant consists of two separate brick buildings. One building is 50 feet by 18 feet and is used to house a complete five-ton experimental refrigerating plant. The other building, which is really the thermal testing room, is 32 feet by 32 feet on the outside, and contains centrally located within its walls the calorimeter room, which is 17 feet by 17 feet. This room is cooled to low temperatures by a number of rows of brine coils set up on three sides. Cold brine is supplied from the refrigerating plant in the adjacent building through 2-inch, heavily insulated pipes. Each of the materials to be tested is made into a cubical box of approximately 65 square feet of outside surface; and this box is then suspended in the calorimeter room for testing. An electric fan and an electric heating coil are placed inside the box for producing the required uniformly distributed temperature difference between the inside and the outside of the box. By this method tests can be made for a very great range of temperature differences. Absolutely constant and uniform temperatures can be maintained for an indefinite period of time, and the

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method of measuring the heat put into the box per unit of time is measured by accurately calibrated voltmeters and ammeters on both the coil and the fan circuits. Direct or "continuous" current is used. Twenty platinum resistance thermometers made by the Leeds & Northrup Company, of Philadelphia, according to special designs for this work, are available for measuring the temperature inside and outside the test-box, as well as in its walls. Cylindrical platinum coils are used for measuring temperatures in all places except on the surfaces of the material where thin platinum resistance disks less than 1/50 inch thick are firmly attached to the surface on their flat sides. Adopting the method proposed by the United States Bureau of Standards, the temperatures have been observed in each test of such materials as are obtainable in blocks of considerable thickness, like cork-board, in the following locations: (1) Two feet from the surface of the material tested, inside and outside; (2) one inch from the surface, inside and outside; (3) on the outside and inside surfaces by absolutely close contact; and (4) at every inch of thickness through the material, with thin, flat platinum resistance thermometers, which can be accurately located.

Some of the most important data obtained during the last year will be presented briefly in this paper in the following order:

1. Variation of heat transmission through glass.
    (a) As affected by humidity,
    (b) As affected by velocity of air over surface,
    (c) As affected by various arrangements for directing the flowing air over the surface tested.
2. Variation of heat transmission through common red brick.
    (a- As affected by humidity,
    (b As affected by velocity.
3. Variation of heat transmission through insulating brick, using high temperatures.

TESTS ON TRANSMISSION OF GLASS

The series of tests studying the effect of heat transmission of glass was made upon a cubical box made of ¼-inch glass, diagramatically shown in Figs. 1-3, and having a mean surface area of 66.67 square feet. The box was placed in the calorimeter room and supported about 2 feet from the floor.

The eectrlica helating coil, together with a small desk fan, was placed on the inside of the glass box, and theremometers placed one inch from the inside surface of the glass, as well as one inch from the outside. The temperature within was controlled by the electrical heating coil, and the air surrounding the glass box was maintained constant by the circulation of the brine in the pipes on the walls of the calorimeter room. The heat delivered to the box and transmitted through the material was measured in the form of electrical energy, and the unit transmission for still air calculated by the following formula:

$$H = \frac{(V_c \times A_c + V_f \times A_f)\ 3.41}{(t_1 - t_2)\ S} \qquad (1)$$

in which

H = unit transmission (B. t. u. per square foot, per hour, per degree difference of temperature measured 1 inch from the surface of the material tested, inside and outside the box.
V_c = voltage of heating coil.
A_c = amperes of heating coil.
V_f = voltage of fan.
A_f = amperes of fan.
3.41 = constant in B. t. u. equivalent for one watt-hour.
t_1 = temperature of air in box, 1 inch from material tested.
t_2 = temperature of air surrounding box, 1 inch from material tested.
S = mean area of test box (66.67 square feet).

Tests showing the variation of the heat transmissions with velocity were run similarly to those previously described, but with the exception that one large fan was used to blow air, which was directed by proper ducts, along one side of the box. This velocity ran from 150 to 1,000 feet per minute and was regulated by dampers suitably located in the ducts. The velocity was measured by means of a carefully calibrated anemometer, averaging the readings for several points along the surface tested.

The heat transmission other than for still-air conditions was calculated as follows:

$$H_2 = \frac{(V_c \times A_c + V_f \times A_f)\ 3.41 - S_1\ (t_1 - t_2)\ H_1}{(t_1 - t_2)\ S_2} \qquad (2)$$

H_1 = unit transmission of tested material under still-air condition (B. t. u. per hour per square foot per degree difference in temperature measured 1 inch from inside and outside surfaces.
H_2 = unit transmission of material tested for velocity.
S_1 = area five sides of test box in still air (55.56 square feet).
S_2 = area material tested (11.11 square feet).
V_c = voltage of coil.
A_c = amperes of coil.
V_f = voltage of fan.
A_f = amperes of fan.
3.41 = constant (B. t. u. equivalent of 1 watt-hour).
t_1 = temperature 1 inch from inside surface. (Average.)
t_2 = temperature 1 inch from outside surface.

It is to be noted in applying formulas for heat transmission with varying velocity that the value of H_1 or the heat transmitted under still-air conditions may have been determined under different conditions of humidity than that for the transmission under varying velocity conditions. It was then necessary, in order to apply this term. to find the effect of humidity on the heat transmission so that the still-air conditions may be corrected to the conditions of the test. The transmission under the given conditions of the test were then corrected to an assumed standard of 80 per cent for comparison.

The varying velocity in the first series of tests upon glass was arranged by terminating the duct from the fan into a small, circular tank or receiver from which small spouts uniformly distributed the air over the surface tested. The arrangement of apparatus used in first series of heat transmission tests is shown in Fig. 1 and results in Table I.

The theoretical disadvantage of this system was due to the possibility of the air flowing over the surface in the form of a jet, and thus producing variable velocities.

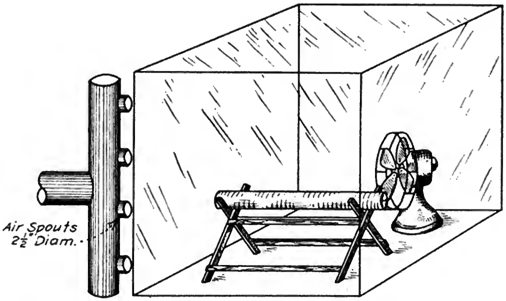

Fig. 1.—Apparatus used in heat transmission tests. First series

Table I.—Results of Tests Showing Effect of Air Velocity upon the Transmission of Heat through Glass. Glass Cube Equipped with Air Spouts. (Fig. 1)

Test No.	Air Velocity. Ft. per Min.	Humidity. %	Unit Trans. Given Conditions.	Unit Trans. Corrected to 80% Humidity.
8	0	77	1.154	1.234
9	400	80	1.640	1.640
10	600	79	1.858	1.883
11	950	79	2.115	2.140

In the second series of tests on glass with varying velocity, the objection to the use of spouts was overcome by arranging the apparatus so that the air would be delivered over the surface in a thin sheet. This was accomplished by terminating the duct into a flattened cone or nozzle, as shown in Fig. 2. The results are given in Table II.

The arrangement of the apparatus used in the second series gave results that were very high as compared with the results as obtained from the first series. The discrepancy was thought to be due

TABLE II.—RESULTS OF TESTS SHOWING EFFECT OF AIR VELOCITY UPON THE TRANSMISSION OF HEAT THROUGH GLASS. GLASS CUBE EQUIPPED WITH AIR CONE, BUT WITHOUT SHIELD. (Fig. 2.)

Test No.	Air Velocity. Ft. per Min.	Humidity. %	Unit Trans. Given Conditions.	Unit Trans. Corrected to 80% Humidity.
12	0	84.4	1.338	1.230
13	350	80.3	4.007	4.000
14	600	75.7	4.427	4.530
15	1025	82.6	4.772	4.708
16	1120	79.6	5.600	5.605

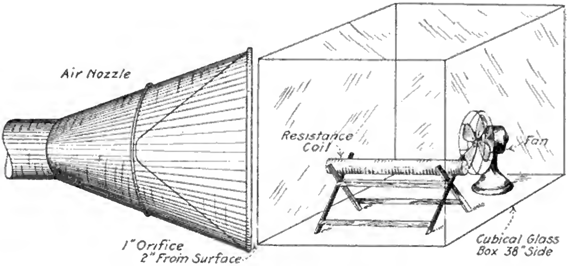

Fig. 2.—Apparatus used in heat transmission tests. Second series.

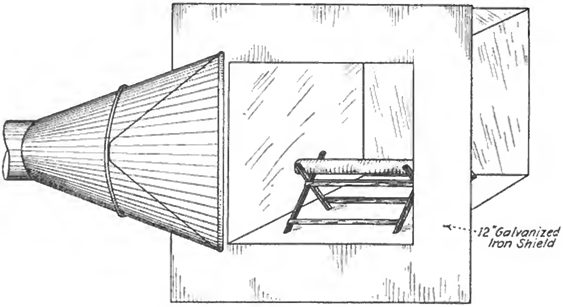

Fig. 3.—Apparatus used in heat transmission tests. Third series

Table III.—Results of Tests Showing Effect of Air Velocity on the Transmission of Heat through Glass. Glass Cube Equipped with Air Cone and 12-Inch Galvanized Iron Shield. (Fig. 3.)

Test No.	Air Velocity. Ft. Per Min.	Humidity. %	Unit Trans. Given Condition	Unit Trans. Corrected to 80% Humidity.
17	0	81.3	1.263	1.230
18	420	74.1	3.409	3.560
19	865	75.8	3.798	3.900
20	1000	79.0	4.010	4.035
21	1000	77.2	3.973	4.040
22	1200	79.5	4.207	4.220

to the production of eddy currents over the surfaces having apparently still-air conditions. In the third series of tests this objection was eliminated by placing a galvanized iron shield 12 inches wide around the surface tested. The results of the tests are given in Table III and the apparatus is shown in Fig. 3.

The results of the tests on the effect of humidity upon the transmission of heat through glass are given in Table IV and graphically shown in Fig. 4.

Table IV.—Results of Tests Showing Effect of Humidity upon the Transmission of Heat through Glass. (Fig. 4.)

Test No.	Air Velocity. Ft. Per Min.	Humidity %	Unit Transmission.
1	0	77	1.154
2	0	79	1.129
3	0	81.2	1.260
4	0	81.3	1.255
5	0	82.6	1.256
6	0	84.3	1.288
7	0	84.4	1.338

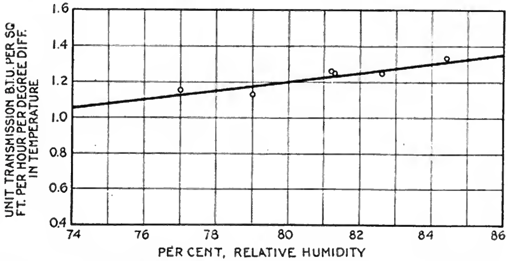

Fig. 4.—Curve showing the effect of moisture in the air on the heat transmission of glass

The results of the tests given in Tables I, II and III are reproduced graphically in Fig. 5.

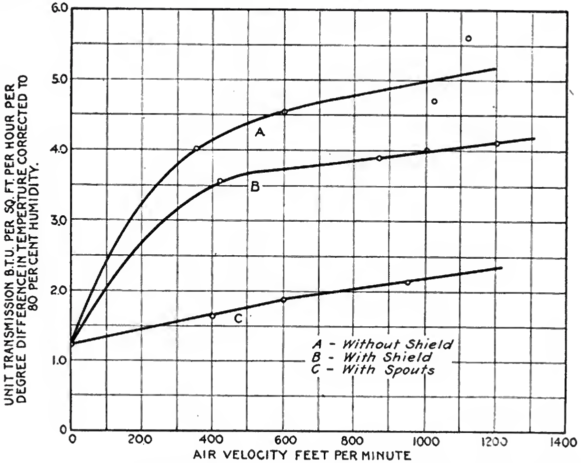

Fig. 5.—Curves showing the effect of air velocity on the heat transmission of glass

TESTS ON THE TRANSMISSION OF COMMON RED BRICK

The study of the transmission of heat through common red brick was performed with apparatus similar to that used in the study of the transmission of heat through glass. A brick cube was constructed of a single thickness of ordinary red building brick laid in approximately ¼-inch lime mortar. The general outside dimensions were 46 inches square, thus making the inside dimensions 38 inches square, as was the case of the glass cube. The heat was supplied to and transmitted through the brick cube, as well as temperature readings, etc., were obtained in the same manner as described for the test made on the glass box.

In the first series of tests the effect of humidity upon the transmission of heat was studied. This was accomplished under still-air conditions and calculated by formula (1), the results of which are shown in Table V.

In the second series the effect of varying the velocity was considered. In this series the apparatus used was identical to that used for glass in the second series of tests, or, in other words, by the use of an air cone or nozzle without a shield. The heat transmission was calculated by equation (2). Results of these tests are given in Table VI.

TABLE V.—RESULTS OF TESTS SHOWING EFFECT OF HUMIDITY ON THE TRANSMISSION OF HEAT THROUGH COMMON RED BRICK. (Fig. 6.)

Test No.	Air Velocity. Ft. Per Min.	Humidity. %	Unit Transmission.
23	0	81	.6645
24	0	83.5	.6986
25	0	83.9	.7094
26	0	84	.7035
27	0	86.5	.7287
28	0	88	.7617
29	0	89	.760
30	0	94	.7780

TABLE VI.—RESULTS OF TESTS SHOWING EFFECT OF AIR VELOCITY ON THE TRANSMISSION OF HEAT THROUGH COMMON RED BRICK. BRICK CUBE EQUIPPED WITH AN AIR CONE, BUT WITHOUT SHIELD. (Fig. 6.)

Test No.	Air Velocity. Ft. per Min.	Humidity. %	Unit Trans. Given Conditions.	Unit Trans. Corrected to 80% Humidity.
31	0	84	.7035	.6635
32	210	83.8	1.157	.9350
33	230	90.5	1.081	.9700
34	450	91.	1.190	1.075
35	660	91.	1.223	1.110
36	660	89.	1.251	1.160
37	765	90.	1.272	1.155
38	850	91.	1.280	1.177
39	1050	91.2	1.348	1.230
40	1100	93.8	1.400	1.250
41	1530	89.	1.420	1.330
42	1530	88.	1.436	1.355

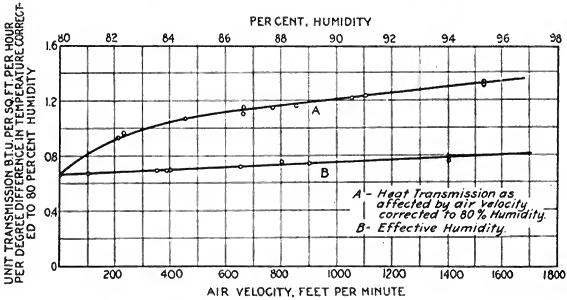

Fig. 6.—Curves showing the effect of air velocity and humidity on the heat transmission of common red brick

Fig. 6 contains curves showing the effect of air velocity and humidity on the heat transmission of common red brick.

TESTS ON THE TRANSMISSION OF HEAT THROUGH INSULATING BRICK

The tests upon the heat transmission of insulating brick were made to study the effect of the transmission of heat through this material under high temperature differences. The unit transmission for this material had never been accurately determined.

The apparatus used in testing the heat transmission of this material was identical with the apparatus used in the tests of common red brick, as previously described. A cube of insulating brick was constructed by a single layer of the bricks, the outside dimensions of which were 46 inches square, thus making the inside dimension 38 inches square, as was the case of the glass cube. The heat supplied to the cube, as well as temperatures, etc., were measured as in other tests. All the tests were run under still-air conditions, the surfaces tested being subject only to the velocity created by natural circulation and the internal temperature maintained inside the cube was varied from 800 to 1,700 degrees Fahrenheit. The results of these tests are given in Table VII and shown graphically in Fig. 7.

TABLE VII.—RESULTS OF TESTS SHOWING EFFECT OF TEMPERATURE IN THE TRANSMISSION OF INSULATING BRICKS. STILL-AIR CONDITIONS. (Fig. 7.)

Test No.	Air Velocity. Ft. Per Min.	Humidity. %	Room Temp. °F.	Box Temp. °F.	Unit Trans.
43	0	45.3	84.7	1695	.2851
44	0	59.3	82.33	1228	.2370
45	0	71.	80.0	830	.1963

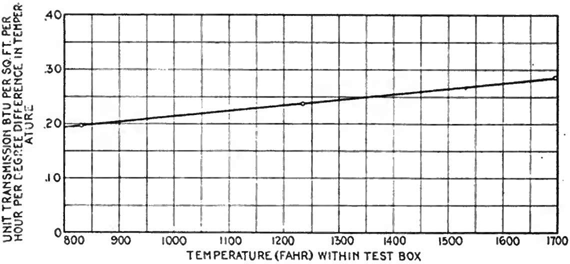

Fig. 7.—Curve showing the effect of temperature on the heat transmission through insulating brick

CONCLUSION

These data show that the transmission through glass and red brick is increased very materially with increase in humidity. Thus for glass the unit transmission was increased 0.25 B.t.u. by increasing the humidity from 74 to 84 per cent, which is equivalent to a change of 24 per cent in transmission for a variation of 10 per cent in relative humidity. For red brick the unit transmission was increased 0.102 B.t.u. by increasing the relative humidity from 80 to 90 per cent, which is equivalent to a change of 15 per cent for a variation of 10 per cent in relative humidity.

The effect of velocity on the transmission of heat, as shown by the data, varies greatly, depending upon the arrangement for conducting the air over the surface tested. In the case of glass the unit transmission corrected to 80 per cent humidity, and at 1,000 feet per minute velocity has these values: 2.19 B.t.u. for a cube equipped with spouts; 4.00 B.t.u. for cube equipped with air cone alone. These factors show very conclusively that in studying the effect of velocity upon the transmission of heat through a material, the arrangement of apparatus for protecting that portion of the material which is considered under still-air conditions is of the utmost importance. It is true, the arrangement of the apparatus as used in the first series of velocity tests, or, in other words, equipping the glass cube with the air spouts, gave values which are low, due to the variable velocities produced by the jets, while, on the other hand, the failure to protect the surfaces not directly acted upon by the moving air gives too high results. In all probability the results with the use of air cone and shield, or as shown in the second series of tests upon glass, give the best results. Assuming this to be true, it is interesting to note that in the case of glass the unit transmission when corrected to 80 per cent humidity gave an increase of 2.78 B.t.u. for an increase of velocity from 0 to 1,000 feet per minute, which is equivalent to an increase of 220 per cent per 1,000 feet per minute of air velocity.

The Engineering Experiment Station, in continuing the study of the effect on heat transmission with varying velocities of air by different arrangements of apparatus, is replacing the glass cube as used in the tests reported here by making five sides of the cube of cork board and the sixth of glass. It is thought that by this arrangement the effect of eddy currents upon the surfaces assumed to be in still air may be practically eliminated owing to the small unit transmission of cork.

The transmission of heat through common red brick was obtained by use of apparatus similar to that shown for glass in the second series of tests or the brick cube was equipped with air cone without shield. These results, judging from the effect of the arrangement as shown in the tests on glass, will be high. But the percentage of error introduced will be small, owing to the low value of the unit transmission of brick. The results of these tests on common red brick when corrected to 80 per cent relative humidity show on increase in unit transmission of 0.56 B.t.u. for an increase of an air velocity from 0 to 1,000 feet per minute, which is equivalent to an increase of 84 per cent for an increase of 1,000 feet per minute in velocity.

These results seem to indicate that in the calculations for the design of heating and refrigerating plants the old method in which an exposure factor is assumed to cover the effect of velocity should be revised.

The increase in heat transmission due to the same velocity is not the same for all materials, and that instead of the use of an assumed constant which is the same for all materials, new values should be obtained for each material. The effect of humidity is another factor which has heretofore been neglected, and, while under ordinary conditions, may not effect the transmissions in a practical sense, yet in installations when the humidity is high this increase in the transmission should not be overlooked.

In the test made upon insulating brick, while the number of tests were small, no definite conclusions can be drawn, but from the results obtained the unit transmission does not remain constant, and varies considerably with the temperature within the test box.

STEAM ECONOMY OF AN ABSORPTION REFRIGERATING MACHINE

By Clarence W. Vogt, Louisville, Ky.

The great development of the internal combustion engine, which has taken place in recent years, gives rise to the question: To what extent will this prime mover supersede steam, our faithful servant of the past?

Because of its comparatively high thermal efficiency, the internal combustion engine using oil for fuel has the advantage in certain fields. With the low price of oil prevailing in the United States, it may become as popular here as is the producer gas engine in Great Britain.

Where power alone is required, and where the heat of evaporation of the steam cannot be reclaimed for useful purposes, the steam plant will be superseded by more efficient power generators, such as hydro-electric or oil engine power.

As refrigerating engineers, we are especially interested in steam, oil or gas from the standpoint of their adaptability to the class of machinery with which we are dealing. Therefore, let us consider the various kinds of fuel in their application to the operation of ice and refrigerating machinery.

Most of us have seen the diagram, in one form or another, showing graphically the small, very small, percentage of "efficiency" representing *heat output* compared with *heat input,* Fig. 1, where only the sensible heat of the steam is utilized in adiabatic expansion within the cylinder of an engine.

If we cannot avail ourselves of the large latent heat component, designated by *C* on the diagram, we must accept a thermal efficiency which is insignificant.

The absorption type of ice and refrigerating machine, however, lends itself admirably to the utilization of this large amount of heat in the exhaust steam. To bring out this point clearly, let us consider a plant of 100 tons ice making capacity per day. The minimum amount of steam necessary for the duty required of such a machine is limited by the steam consumption of the ammonia generator. For a conservative basis of figuring, this may be taken at 60 lbs. per hour per ton of ice making capacity.

In order to prove that this quantity of 60 lbs. is a liberal allow-

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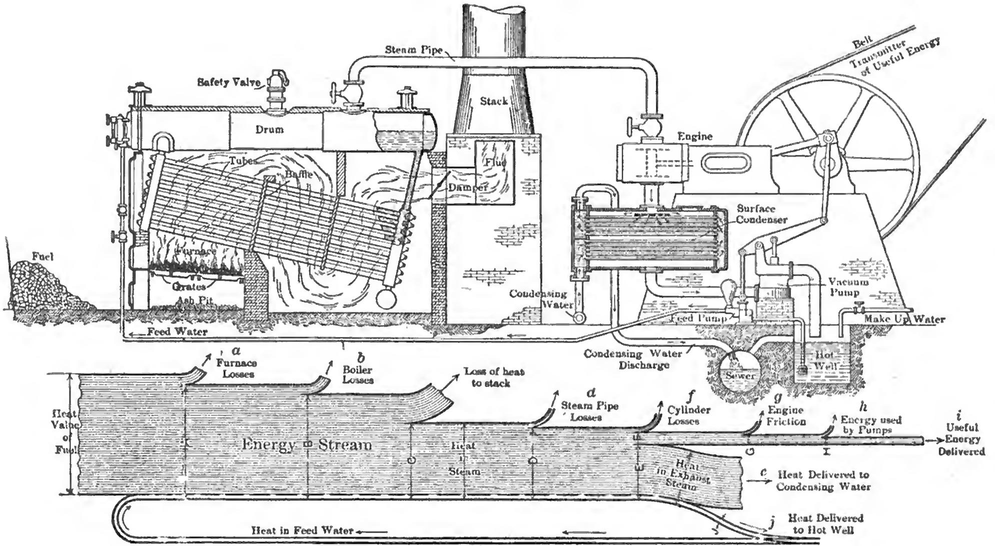

Fig. 1.—Elements of a steam-power plant

ance, Table I is submitted, based on a log sheet taken from a recent test of a 40-ton absorption machine.

You will note from this test that the steam condensed per ton of refrigeration was 28.62 lbs. per hour by actual weight. Also, that the temperature of the cooling water supplied was never lower than 75 degrees F., and the head pressure 150 lbs.

Furthermore, the temperature of the brine was +6 degrees F., which is five or six degrees lower than the temperature maintained in the average ice tank. These two conditions tend to increase the steam consumption over the amount that would be necessary with 60 degrees cooling water, and eleven or twelve degree brine.

To ascertain the amount of condensation per ton of ice making capacity, it is only necessary to double the consumption mentioned, although some authorities figure only 1¾ tons refrigeration per ton of ice making capacity. Hence, we have $28.62 \times 2 = 57.24$ lbs. of condensation. Evidently our figure of 60 lbs. of steam per ton of ice making capacity is conservative, and larger than the amount actually required in plants where the exchanger and other parts are properly proportioned. I mention the exchanger in this connection, because when its surface is insufficient, the rich liquor enters the generator with a correspondingly lower temperature, so that additional heat is required to bring it to the temperature at which the gas is driven off.

Sixty pounds of steam per hour means two boiler horsepower per ton of ice making capacity, or 200 boiler horsepower for a 100 ton plant, provided the plant is equipped to make ice from raw water. This is the only fair comparison to make, because with distilled water ice the oil or gas engine is at a great disadvantage.

The purpose of this paper is to show that, although the average engineer, without second thought, concedes the raw water ice making field as naturally belonging to the compression type of machine, this concession is not justified, and therefore the properly designed absorption machine may be expected to successfully enter this field in competition with compression machines.

Returning to our figures we find that the amount of steam required is 6,000 lbs. per hour, or 200 boiler horsepower capacity, while a compression machine of equal capacity, if driven by a simple Corliss engine, requires 300 boiler horsepower. With the absorption machine no boiler feed water heater is needed, nor even a boiler feed pump, provided a return trap is used for returning condensation to the boilers.

The auxiliaries in an ice plant require from ½ to ¾ brake horsepower per ton of ice making capacity. The steam consumption of these auxiliaries, when working against a back pressure of 15 lbs., will be materially less than the amount required by the ammonia generator, which, as stated, will not exceed 6,000 lbs. per hour, and, therefore, the auxiliaries may use on an average 100 lbs. of steam per brake horsepower per hour and still no steam will be wasted.

TALBE I
DAILY REPORT OF TEST OF ABSORPTION MACHINE. AUGUST 27, 1915.

Time.	PRESSURE, LBS. PER SQ. INCH.				Ammonia Pump R.P.M.	Brine Pumped Gals. Per Min.	Water Pumped Gals. Per Min.	TEMPERATURES IN DEGREES FAHRENHEIT.						
	Generator Steam.	Generator Ammonia.	Absorber.	Brine Cooler.				Brine to Cooler.	Brine from Cooler.	Brine Range.	Water from Condenser to Absorber.	Water from W. L. Cooler.	Water from Absorber.	Cooling Water.
11.00 A.M..........	8¾	150	13½	15¼	18	198	165	11¼	5¾	5½	82	107	90	75
.15 "	8½	150	14	15¾	18	200	165	11¾	6¼	5½	..	...	..	75
.30 "	8½	150	14	15¾	18	200	162	11¾	6¼	5½	81	106	90	75
.45 "	8¾	150	14	15¾	18	202	168	12	6½	5½	..	...	..	75
12.00 M............	9	152	14	15¾	18	205	165	12	6½	5½	82	106	90	76
12.15 P.M..........	9	150	14	15¾	18	205	170	11¾	6½	5¼	..	...	..	76
.30 "	8¾	150	14	16	18	200	175	11¾	6¼	5½	81	104	90	75
.45 "	8¾	150	14	15¾	17	205	173	11¾	6¼	5½	..	...	..	75
1.00 P.M..........	8¾	150	14	15¾	18	205	170	11¾	6½	5¼	81	105	90	75
.15 "	8¾	150	14	15¾	18	205	170	11¾	6¼	5½	..	...	..	76
.30 "	8¾	150	14	16	18	207	170	12	6¾	5¼	81	107	90	75
.45 "	9	150	14	16	18	205	168	12	6¾	5¼	..	...	..	75
2.00 P.M..........	8¼	149	14	16	17	205	168.	12	7	5	81	103	90	75
.15 "	8¾	151	14	15¾	18	208	170	11¾	6½	5¼	..	...	..	76
.30 "	8½	150	14	15¾	18	205	165	11¼	6	5¼	83	108	90½	75
.45 "	8½	150	14	15¾	18	202	170	11¼	6	5¼	..	...	..	75
3.00 P.M..........	8½	150	14	15¾	18	200	170	11¼	6	5¼	82	108	90	75
.15 "	9	150	14	15½	18	202	170	11½	6	5½	..	...	..	75
.30 "	9	150	13¾	15¼	18	205	173	11½	6	5½	81	105	90	75
.45 "	9	150	13¾	15¼	18	212	173	11½	6	5½	81	103	90	75
4.00 P.M..........	8½	150	13¾	15¼	18	203	173	11½	6¼	5¼	81	104	90	76
.15 "	8½	150	13¾	15¼	18	205	173	11½	6¼	5¼	..	...	..	75
.30 "	8½	150	13¾	15½	18	203	170	11¼	6	5¼	81	103	90	75
.45 "	8½	150	13¾	15½	18	203	170	11¼	6	5¼	..	...	..	75

TABLE. I (Continued.

TOTAL METER READINGS AND AVERAGES PER HOUR.

TIME.	11.00 A.M.	12.00 M.	Average.	1.00 P.M.	Average.	2.00 P.M.	Average.	3.00 P.M.	Average.	4.00 P.M.	Average.
Refrigeration in tons........	....	....	45.3	....	45.0	....	....	....	....	....	43.7
Brine metered Gallons.......	3,529,730	3,542,080	206.0	3,554,520	207.3	3,567,600	218.0	3,579,763	201.6	3,591,862	201.6
Water metered, Gallons......	4,203,330	4,213,400	167.8	4,223,725	170.0	4,234,250	175.4	4,244,110	168.3	4,254,210	168.3
Gals. water per minute per ton of refigeration		....	3.7	....	3.8	....	3.8	....	3.8	....	3.8

CONDENSATION FROM AMMONIA GENERATOR.

From 1.00 P.M. to 1.05 P.M.	106 lbs.	From 3.00 P.M. to 3.05 P.M.	110 lbs.
1.10 " " 1.15 "	110 "	3.10 " " 3.15 "	111 "
1.20 " " 1.25 "	112 "	3.20 " " 3.25 "	105 "
1.30 " " 1.35 "	110 "	3.30 " " 3.35 "	109 "
1.40 " " 1.45 "	108 "	3.40 " " 3.45 "	105 "
1.50 " " 1.55 "	100 "	3.50 " " 3.55 "	102 "
Fo thirty minutes	646 lbs.	For 30 minutes	642 lbs.

Total 1,288 lbs. per hour Average amount of steam condensed, 28.62 pounds per hour per ton of refrigeration.

TABLE II

COST OF MANUFACTURING RAW-WATER ICE—100 TONS CAPACITY—VARIOUS SOURCES OF POWER

(*From General Electric Review, July,* 1914)

	Motive Power	Using	h.p.	Total h.p.—per hr. per Day	Fuel per h.p.	Fuel or Power Per Day	Unit Cost of Fuel	Fuel Cost per Day	Tons of Ice per Unit of Fuel	Sundry Costs			Labor Cost					Total Cost per Day	Cost per h.p. of Power Plant
										Oil Waste. Etc.	Ammonia	Calcium	1 Day Engineer	1 Night Engineer	4 Ice Pullers	1 Extra Man	2 Firemen		
1	Simple Steam Engine	Bituminous Coal	367 i.h.p.	8808	3½lb.	15.4 tons	$2.00/ton 2.50/ton 3.00/ton	$30.80 38.50 46.00	6.38 Tons Ice per Ton of Coal	2.00	2.50	0.50	5.00	3.25	8.00	2.00	5.00	$59.05 66.75 74.45	$32.00
2	Compound Condensing Engine	Bituminous Coal	375 i.h.p.	8952	2 lb.	8.95 tons	2.00/ton 2.50/ton 3.00/ton	17.90 22.38 26.85	11.15	3.50	2.50	0.50	5.00	3.25	8.00	2.00	5.00	46.15 50.63 55.10	40.00
3	Producer Gas Engine	Bituminous Coal	360 b.h.p.	8640	1¼lb.	5.4 tons	2.00/ton 2.50/ton 3.00/ton	10.80 13.50 16.20	18.5	3.50	2.50	0.50	6.00	4.00	8.00	2.00	4.00	*41.30 44.00 46.70	65.00
4	Producer Gas Engine	Anthracite Coal	360 b.h.p.	8640	1 lb.	4.31 tons	3.00/ton 3.50/ton 4.00/ton 4.50/ton 5.00/ton 5.50/ton 6.00/ton	12.93 15.09 17.24 19.40 21.55 23.71 25.86	23.2	3.50	2.50	0.50	6.00	4.00	8.00	2.00	4.00	43.43 45.59 47.74 49.90 52.05 54.21 56.36	65.00
5	Producer Gas Engine	Lignite	360 b.h.p.	8640	1¾lb.	7.56 tons	2.00/ton 2.50/ton 3.00/ton	15.12 18.90 22.68	13.2	3.50	2.50	0.50	6.00	4.00	8.00	2.00	4.00	45.62 49.20 53.18	65.00

TABLE II—(Continued)

COST OF MANUFACTURING RAW-WATER ICE—100 TONS CAPACITY—VARIOUS SOURCES OF POWER

(*From General Electric Review, July,* 1914)

	Motive Power	Using	h.p.	Total h.p.-per hr. per Day	Fuel per h.p.	Fuel or Power Per Day	Unit Cost of Fuel	Fuel Cost per Day	Tons of Ice per Unit of Fuel	Sundry Costs			Labor Cost					Total Cost per Day	Cost per h.p. of Power Plant
										Oil Waste, Etc.	Ammonia	Calcium	1 Day Engineer	1 Night Engineer	4 Ice Pullers	1 Extra Man	2 Firemen		
6	Gas Engine	Natural Gas 1050 B.t.u. Value	360 b.h.p.	8640	10 cu.ft.	86400 cu. ft.	$0.20 per M 0.25 per M 0.30 per M. 0.35 per M	$17.28 21.60 25.92 30.24	864 cu. ft. gas req'd per Ton of Ice produced	3.50	2.50	0.50	6.00	4.00	8.00	2.00		$43.78 48.10 52.42 56.74	$35.00
7	Oil Engine	Fuel Oil	360 b.h.p.	8640	0.075 gal.	648 gal.	$0.02½ gal. 0.03 gal. 0.03½ gal. 0.04 gal.	16.20 19.44 22.68 25.92	0.155 Tons Ice per Gal. of Oil	3.50	2.50	0.50	6.00	4.00	8.00	2.00	4.00	46.70 49.94 53.18 56.42	40.00 to 60.00
8	Motor	Electricity	327 on wire	7848	0.746 kw.	5854 kw.	$0.00½ kw. 0.00⅝ kw. 0.00¾ kw. .000⅞ kw. 0.01 kw.	29.27 36.48 43.90 51.23 58.54	58.54 kw-hr req'd per Ton of Ice produced	2.00	2.50	0.50	5.00	3.25	8.00	2.00		47.51 59.63 67.15 74.48 †81.79	16.00
9	Exhaust Steam Absorp. Machine	Bituminous Coal	200 boiler h.p.	4800 boiler h.p.	3¾lb	9 tons	$2.00/ton 2.50/ton 3.00/ton	18.00 22.50 27.00	11.1 Tons* Ice per Ton of Coal	1.00	2.50	0.50	5.00	3.25	8.00	2.00	5.00	45.25 49.75 54.25	30.00 per boiler h.p.

Water estimated at 70 deg. Fahr. Suction pressure 15 pounds. Power included for pumping water with 60-foot head. * Minimum. Condenser pressure 185 pounds. Ammonia used as refrigerant. † Maximum. Above costs do not include interest on investment, depreciation, taxes, insurance, management, or any other fixed charges. Power is based on summer conditions. Less power will be required if yearly average is figured on. Oil engine fuel requirements based on engines of Diesel type.

Table II* is a reprint of the one found in the A. S. R. E. JOURNAL, July, 1915, pages 46, 47, with one column, 9, added at the bottom. The figures given in this table are only approximate, but I am sure they are fair and not random computations. You will note in column 9 that the cost of operating the absorption machine per day, excluding interest on investment or depreciation, compares favorably with the other types shown in the table. The figures given in column 9 may be accepted as conservative, based on tests and records taken from actual operation.

The investment from which interest and depreciation must be figured will be seen from the end column to come well under the cost of any of the other power sources showing a correspondingly low cost of operation.

These two items must be considered together in order to arrive at a decision as to the most profitable investment. For example, let us compare the absorption machine using coal at $2.00 per ton, with the oil engine using oil at 3 cents a gallon. The cost per day, exclusive of interest on investment and depreciation, will be practically equal. But when we consider that the oil engine power plant will cost $12,000 more than the steam plant of the absorption machine, it is easily seen that the added investment necessary could never show a profit. Again, drawing a comparison between the absorption machine and an electrically-driven compressor; where coal costs $2.50 per ton, and electricity is purchased at ¾ cent per kilowatt hour, it will be seen that its operating cost will be $17.40 less per day, exclusive of interest and depreciation. Since the cost per horsepower of the power plant multiplied by the number of horsepower required in each case is almost equal, showing less than $800 in favor of the electrically-driven plant, this added investment is paid for by the decrease in operating expenses at the rate of over 300 per cent per year, figuring 150 days operation, thus $17.50 × 150 = $2,610, which is the difference in cost of operation per year. On this basis the absorption machine would be more profitable, even if its first cost would exceed that of the electrically-driven compression machine by $13,000, although, as stated, the first cost of the two will be almost equal. This not only holds good in the two cases cited, but in the case of each type shown.

With the advent of raw water ice, the absorption machine will not be limited to refrigerating work—it will continue to compete in ice making duty as heretofore, because of its economy and profit-producing power.

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* Table II originally was prepared by the Arctic Ice Machine Company, Canton, Ohio, as stated in The Journal, September, 1915, page 80, and published in the *General Electric Review*.

THE QUINCY MARKET SAFETY VALVE FOR AMMONIA COMPRESSORS

By F. L. Fairbanks, Boston, Mass.

While assisting the Massachusetts Board of Boiler Rules in its endeavor to find a reliable safety valve for ammonia compressors, the writer had occasion to perfect a valve which was already in use at the plant of the Quincy Market Cold Storage & Warehouse Co., Boston, Mass.

Tests were made at the plant mentioned of six different makes of ammonia safety valves under various conditions and pressures. Most of these tests were witnessed by members of the Board of Boiler Rules and by members of the A. S. R. E. Committee on Regulations for Refrigerating Plants, appointed for this purpose.

The conclusions drawn from these tests were as follows:

1.—That valves with iron or steel seats were not safe, for the reason that the disc is readily corroded to its seat by a small amount of moisture, two of the six valves tested failing to open at 350 to 375 lbs., although they were set at 250 lbs.

These valves upon being dissembled were found to be rusted solidly to their seats and to the winged guide in the port; this was undoubtedly due to the small amount of moisture left by the steam which was used by the maker to test the valve and set the spring, notwithstanding the fact that the rust was perfectly dry when the valve was taken apart.

2.—That the cast iron valve discs, with winged guides in the port, would not stand well the excessive chattering and hammering on its seat, to which most of the valves were subject when opening and closing, this being due to the surging of gas pressure which is always present in the discharge line of a compressor, especially at higher speeds.

One of the valves broke off its winged guide in the port and wedged the disc, so that the line had to be shut off. Examination showed that the metal of both seat and guide, which were of cast iron, were of good material and sound, the fracture being caused by the heavy chatter and pound as it opened and returned to its seat.

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3.—That the capacity of all the valves, except the pop valve, was very low.

4.—That the pop valve gave practically no warning when it was about to blow, and for this reason would be dangerous to anyone who might be in the path of the discharge when it opened. Also, that the seat of the pop valve could not stand the blow of the valve when closing, and remain tight.

5.—That none of the valves, notwithstanding their low capacity, could discharge through a pipe line no larger than their discharge outlet and still maintain their setting and character.

It was shown that a 2-inch valve would require a 6-inch pipe connection to carry its discharge gas the first 100 feet, with a proportionate increase for each added length, and this without taking into consideration bends or fittings which would be necessary in most cases. Discharge pipe lines with test gauges were erected, so that this feature could be definitely demonstrated, the line being so constructed that readings could be taken for both 50 feet and 100 feet.

The regulations formulated were based upon the knowledge gained from these tests, and it was then found that to comply with the regulations an entirely different form of safety valve from any obtainable would have to be designed.

The writer was selected to do this work, in as much as the valve then in use in our plants required only minor changes to enable it to meet the conditions specified. This has been done, the valve thoroughly tested out, and it has been accepted.

DESCRIPTION OF THE VALVE

The Quincy Market safety valve is shown in Fig. 1. It has no guide in the port, being guided by a bushing pressed into the valve body and by a cylindrical piston which is a part of what would be called the valve disc.

This bushing is bored and reamed after being pressed in, and all alignment of the valve depends upon this bushing, the seat being reamed through it as a guide, thereby making all parts of valve body and bonnet self-centering and self-aligning.

The valve is cone pointed to present the least possible resistance to the flow of gas, and is turned to an angle of 60 degrees, the seat being reamed 1/32 inch wide and the angle being 57 degrees, this allowing the valve to seat on an edge and not on a flat surface.

The piston guide is of sufficient excess area to secure the characteristics called for in the regulations, and to cause the valve to open wider as the gas discharge increases and the back pressure builds up; the construction being such that the valve has these same features whether it is discharging through 500 feet of pipe or directly from the valve outlet.

It has the very desirable feature for ammonia work, of opening at any predetermined point, the same as an old-fashioned lever safety valve, giving the operator sufficient warning to enable him to remedy the trouble in many cases; when this point is reached the

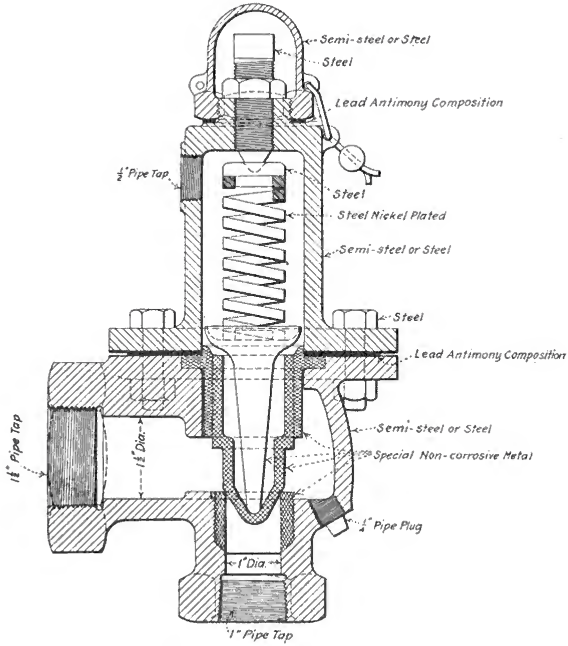

Fig. 1.—The Quincy Market ammonia compressor safety valve

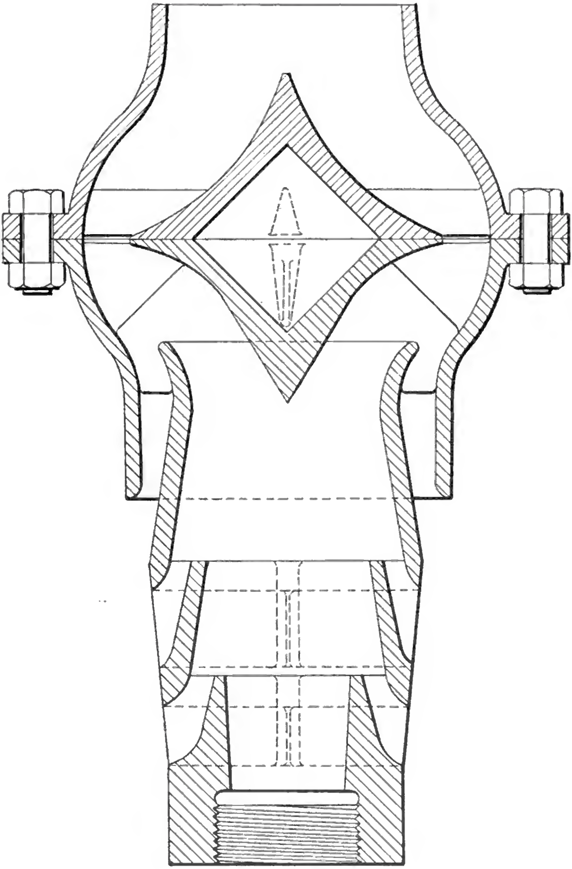

Fig. 2.—Ammonia diffuser

valve continues to open as the pressure on the discharge side backs up on the increased area of the piston guide, the entire range being gradual and continuous until the back pressure reaches 0.585 of the initial pressure, thus giving in a valve practically the capacity and character of a Napier orifice.

The return movement is like the opening movement, the valve leaving and returning to its seat without chatter or shock of any kind.

The non-corrosive metal used was originated by the writer, and is the same as has been in service in steam and ammonia valves at the plant of the Quincy Market Cold Storage & Warehouse Company for over eight years. It is not affected by any of the common corrosive elements and is very hard—so hard, in fact, that it requires grinding to get a good surface.

The capacity of this valve was demonstrated by tests in which the gas from two 26-inch by 48-inch single-acting cylinders at 60 revolutions per minute was discharged through 100 feet of 1½-inch pipe having seven elbows; the initial pressure being 272 lbs., and the back pressure on the discharge outlet of the valve only 130 lbs. per square inch gauge, or, in other words, the valve handled easily a displacement of 1,700 cubic feet per minute, with 30 lbs. to spare on the outlet side of the valve, and this without chatter, shock or any other sign of distress.

DESCRIPTION OF DIFFUSER

The inlet to the nozzle is to be of the same diameter as the outlet of the safety valve. The diffuser shall be attached in a vertical position and made to prevent obstruction of its openings by rain, snow, ice or otherwise, and not likely to be seriously corroded by the gases or liquids with which it comes in contact.

The type of diffuser illustrated by Fig. 2 embodies the principles necessary. It will be seen to be a simple gas ejector, mixing the issuing ammonia gas with a large proportion of air, so as to dilute and break it up thoroughly, with no chance of obstruction, and discharge upward. It is made of cast iron in the various sizes called for in the regulations.

THE RATE OF HEAT TRANSFER IN DOUBLE-PIPE BRINE COOLERS*

By Fred Ophuls and Van R. H. Greene, New York, N. Y.

ABSTRACT AND DISCUSSION

The object of the paper is to present a method of analyzing brine cooler performance, and work out data of sufficient accuracy to permit of designing double-pipe brine coolers for any desired brine temperature range.

A brief description of the test plant is given, also a diagrammatic sketch showing its arrangement.

It is pointed out that, in designing these coolers, the resistance offered to the brine flowing through them must be taken into consideration. Before making the test runs for obtaining data on the rate of heat transfer, tests were made to determine the change in the brine friction with the varying quantities of the brine pumped through such coils. Tables 1 and 2 give the results of all these tests. Plates 1 and 2 show the curves found. To make this data available for designers, a new constant to represent these test results was found for Arthur J. Martin's formula giving the friction of fluids in pipes. The final equation 3 can be safely used to determine the friction losses in flat pipe coils, similar to those of the coolers tested.

The subject of heat transfer is introduced by a discussion of the phenomena producing a change in the heat transfer with a change in the fluid speeds and of the complications which arise when analyzing such test results. The basic formula, which states that the rate of heat transfer is equal to a constant times the square root of fluid velocity is used to analyze the results.

The paper gives in detail all average readings of the tests used in the calculation (see Table 3). The method adopted in figuring the rate of heat transfer is shown in Table 4. It was found that, when plotting the rate of heat transfer against the speed of the brine in the coolers, no satisfactory curve could be found which would represent the results obtained. A similar plot using the final speed of the ammonia vapor in the cooler was not productive of better results. This led the authors to believe that the rate of heat transfer varied not only with the speed of the brine, but also with the mean speed of the ammonia vapor in the coolers.

To obtain the mean speed of the ammonia vapor in the coolers, the increase in vapor speed in the annular space was found from the decrease in the temperature of the brine as it passed from one pipe to the other. Readings of these temperatures had been taken at various times. From these data it was possible to express with some accuracy the variation of vapor speed in the form of equation 6.

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* Read at Fourth Western Meeting, San Francisco, Cal., September 23 and 24, 1915. Complete paper appeared in The Journal, September, 1915.

By integrating this equation between proper limits the arithmetic mean vapor speed is found. The rate of heat transfer is plotted against the mean speed of the ammonia vapor plus the speed of the brine. The results so obtained are not satisfactory, as will be seen from curve and points in Plate 6.

If the form of the basic equation is correct, the mean speed of the ammonia vapor which should be used in this analysis would be the mean square root speed. It was found, however, that the value of the mean square root speed differed but slightly from the arithmetical mean speed, and the results of the new plot shown on Plate 7 required further analysis. For this purpose the temperature of the ammonia at various points in the annular space of the coolers was available, and it was found that the lack of uniformity in the location of the points plotted was not due to the inaccuracy of the basic equation, but due to the fact that in a great many of the tests an insufficient supply of liquid ammonia had been fed to the coils, so that a large part of the cooling surface was not effectively used. By correcting the test results in view of this information, as well as could be done, it was found that very much better plots were obtained. However, the points shown on these plots (see Plate 9) do not cover a wide enough range of the heat transfer coefficient. Further tests should be made with greater variations in brine and ammonia speeds and tonnage. The final equation 10 is therefore only approximate.

The insufficient supply of liquid ammonia to the coolers was caused by the constant liquid head maintained by the accumulator attached to the coolers. For making such tests, hand operated expansion valves should be used, and if the effect of the flooded system on the brine coolers of this type is to be determined, a liquid cooler operated by a separate expansion valve should be used, or an accumulator installed whose height above the coolers can be varied to suit the load on the coolers.

The conclusions arrived at are, that the basic equation can be safely used in this connection, and that for high temperature ranges of the brine it is necessary to use several banks of coolers through which the brine flows in series and the ammonia liquid in parallel.

DISCUSSION

Henry Torrance.—This is a very interesting paper. The authors are well versed in the laws relating to heat transmission. They have developed an excellent formula, but before applying it one must be sure that the engineer will manipulate the expansion valves to suit the thermometer readings, *taken at each return bend of each brine cooler,* so the vapor will leave just in a saturated condition. The company I am with has installed many shell type brine coolers, but if it had to first calculate the capacity by means of a formula, such as the one proposed, very few would have been installed by it.

It seems to me that this paper nails the lid on the coffin of the double-pipe type of brine cooler. It is expensive in first cost, extravagant in power consumption, clumsy, and difficult to operate.

The best result, apparently, was obtained at test No. 502, when the mean temperature was 5.8 degrees F. The surface required was 29 square feet per ton. A shell cooler under the same conditions would require less than one-half this amount of surface.

In their ingenious equation 6, the authors have devised a formula for calculating the capacity of the cooler at any point by means of the everchanging velocity of the evaporated ammonia. What a task!

Double-pipe coolers of the usual construction are imperfect, because the annular space between the two pipes is too small to allow the evaporated liquid to pass through without priming. Boilers are evaporators. Who ever heard of a double-pipe steam boiler or water evaporator? A large vapor liberating surface is required, and therefore *we should use shell type brine coolers,* enabling us to work with any desired brine velocity and still have the coils submerged in liquid. Then we will not require that equation 6, $y^{1.3} = a\ x$, and can dispense with so many thermometers.

So far as I know, none of the consulting engineers among the members of our Society recommend double-pipe brine coolers, and I cannot understand why so many of them are being installed by people who should know better, except that they are easily sold and appear to be inexpensive. In realty they are expensive, because about half of their surface is of little use, unless, as recommended in the paper, a thermometer is used on each return bend.

The first shell type brine coolers were manufactured by Pontifex and Wood, of England, for their absorption machines. One of the first engineers in this country to recognize their value was George H. Stoddard, of the Quincy Market & Cold Storage Co., Boston, Mass. These coolers had a vertical shell containing extra heavy helical pipe coils through which the brine was pumped. They can be operated horizontally as well, but the coil surface above the liquid level is then of little value. Used vertically, the rate of heat transmission is over double that shown in the best tests given in the paper.

If I had to pay for the coal and labor on a plant, I would not take a double-pipe cooler as a gift.

I compliment the authors for their good work done on an apparatus which, however, is not worth it.

Van R. H. Greene.—Mr. Torrance was kind enough to show me his comments before my paper was read, and for that reason I was on my guard and said, by way of introduction, that our paper was not an exposé of superiority of the double-pipe brine cooler, but that our object was simply to offer a means for finding the rate of heat transfer in that type of apparatus. The formula arrived at applies to atmospheric and double-pipe ammonia condensers, as well as to double-pipe brine coolers.

I agree with Mr. Torrance that the efficiency of the double pipe brine cooler is not equal to that of the shell type of cooler, for the reason that the friction of the gas in the annular space is so great that only a moderate temperature difference can be obtained, which limits the rate of heat transfer. As to best method of operation, I do not think it makes any difference whether a double pipe or a shell type cooler is used. The only way they can be operated efficiently is with a thermometer in the outlet. By this means it is possible to see whether the gas leaving the cooler is saturated as it should be. If it is, one can be sure that all other pipes throughout the cooler lengths contain liquid ammonia.

Mr. Torrance also was kind enough to show me a telegram received from his company, which informed him that tests conducted by his company on a shell type brine cooler have shown a rate of heat transfer running as high as 140 to 185 B.t.u. But they also found quite a few other tests where the rate of heat transfer was no better than in a double-pipe brine cooler. This they attribute to the fact that the rate of liquid feed was not enough to wet the whole surface, which is liable to happen when there is no thermometer in the gas outlet pipe.

G. H. Geisler.—Has any one made experiments on the friction of water through a double-pipe condenser? I have found that by reducing the number of pipes per stream I can circulate the water with a much lower head. For example, when using only one water circuit on a double-pipe (1¼ and 2 inch) condenser 12 pipes high, almost a two-stage centrifugal pump is necessary to get sufficient water through, while with 6 pipes only per stream, a single stage pump is sufficient.

Henry Torrance.—The company I am connected with always arranges for a double flow on such condensers, two streams of cooling water passing in parallel through the two sets of six 1¼-inch pipes. Very interesting data on friction in brine coolers is given by Messrs. Ophuls and Greene in their paper.

THE OIL ENGINE FOR ICE PLANT SERVICE*

By Louis K. Doelling, New York, N. Y.

ABSTRACT AND DISCUSSION

Internal combustion engines in the past ten years have been so perfected that they now successfully compete with steam engines in all classes of work, but it is only within the last three years that they have been introduced on a large scale in refrigerating plants. The hindrance to their employment for this service was formerly the need of large quantities of steam in such plants, but through change in the design of the plants themselves, so much steam is no longer required.

Figures are given to show the high thermal efficiency of the oil engine, which is demonstrated to be two and a half times that of the steam engine. Oil engines used in refrigerating plants may be divided into four classes:

(1) High compression or Diesel type engines.
(2) Medium compression engines.
(3) Four-cycle low compression engines.
(4) Two-cycle crankcase scavenging and one-to-one scavenging engines.

The working processes of these various classes of engines are described, and the commercial designs on the American market are illustrated. Comparison is made in some detail of the fuel consumption of the various types, and also the grades of oil on which they operate satisfactorily.

When installing an oil engine to drive a refrigerating machine, it should be remembered that the oil engine differs from the steam engine in several respects. The most important difference is the impossibility of overloading the oil engine appreciably for long periods. Oil engines usually run at such a speed that the compressor cannot be directly coupled, but must be connected through a belt or rope drive. Transmission equipment to connect oil engines and refrigerating machines must be of a very heavy design.

The water supplied for cooling the jackets of the oil engine must be sufficient in quantity and must not form scale. Where the natural water available is not suitable, a simple re-cooling system should be installed, using rain water or distilled water.

The exhaust piping for the oil engine should be arranged in a careful manner and a summary is given of the points to be considered. The recovery of heat from the oil engine jacket water is quite feasible, but the installation of a heater to obtain some of the heat in the exhaust gases has not usually proved a commercial success.

A great advantage of an oil-engine-driven plant is the simple manner in which the fuel is handled and the absence of ashes and clinkers and their accompanying troubles.

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*Read at Fourth Western Meeting, San Francisco, Cal., September 23 and 24, 1915. Complete paper appeared in The Journal, September, 1915.

Oil engine lubrication should have the same attention as that of a high speed steam engine. Characteristics are given of suitable oils for cylinder and bearing lubrication. The lubrication system adopted by the De La Vergne Machine Company, including their oil reclaimer, is described.

In installing oil engines for ice making plants, the following recommendations are made:

(1) Calculate the horsepower required for the compressor on the following assumptions: Back pressure, 20 lbs; head pressure, 220 lbs; exponent of compression curve, 1.3; mechanical efficiency of compressor, 85%; efficiency of belt or rope drive, 85%. Auxiliaries, if driven electrically, will require from 1 to 1¼ horsepower per ton of ice for a can plant.

(2) - The compressor and auxiliaries should be driven from separate engines.

(3) More than one unit should be installed.

Detailed tables are given showing a comparison of the investments required with oil engine and steam-driven ice making plants, also a comparison of the daily and yearly costs of operation.

High-speed compressors are now being developed for direct connection to oil engines. Such a unit has many advantages in floor space, first cost, increased efficiency, reduced fuel consumption and reduced cost of maintenance. As soon as the high speed ammonia compressor has proved a success in commercial use, it will certainly be employed generally, because the cost of an oil engine driven unit will be less than that of a steam plant.

The experience gained in the manufacture of ice under commercial conditions during the last four years makes it possible to install, at reasonable cost, oil engine driven plants which will give perfect satisfaction.

DISCUSSION

G. H. Geisler.—What gravity of California fuel oil, Mr. President, do you use with your engine? We have an asphaltum base oil on the Coast which I understand is difficult to burn in the Diesel type of engine.

President Doelling.—The lowest is about 12 degrees Baumé, (specific gravity, 0.986). There are some oils which you can get here of about 16 to 18 degrees gravity.

G. H. Geisler.—Can you burn them successfully?

President Doelling.—Yes. With most of these heavy oils it is only necessary to have a very good filter, because they contain a lot of fine dust, mostly limestone and sometimes carbon, which act on the cylinder of the engine like emery. Instances are known where a cylinder was worn out in four weeks. Crude oil is difficult to obtain. Besides, it is more expensive than distillate, because it contains the gasoline and lubricating oils. If you ask for a fuel oil, you get a distillate. Therefore, manufacturers of oil engines are agreed that it is not necessary to make any specifications, except that there shall not be more than one to one and one-half per cent. of water in the oil. Sometimes the water is twenty-five per cent., forming an emulsion which cannot be used in an engine, because the water is not combustible.

G. H. Geisler.—I understand that the Union Ice Company is using one of your oil engines over in Oakland. They use oil of 16 degrees Baumé, while the Diesel engines of the Allis-Chalmers exhibit at the Exposition are supplied with oil of 24.5 degrees.

President Doelling.—Yes. In Diesel engines a distillate oil is preferred, a heavier oil being more difficult to ignite and necessitating a higher degree of compression. A type of engine, similar to ours, is being built extensively in England for export to the Far East, where they have very heavy oils. The oil is easily ignited on plates that are red hot, thus the heavy oil is successfully burnt in those engines, but in this country this is no great advantage, because here the distillate oil may be obtained almost as cheaply as the heavy oil.

J. N. Gruhl.—If two oil engines have two 6-inch exhaust pipes combining into one main, is it necessary to make the area of the main equal to the combined area of the two pipes?

President Doelling.—Yes, it is advisable to do that. But the two small pipes should be brought together in such a way that one exhaust does not interfere with the other, so that you would get a suction effect.

J. N. Gruhl.—What size main would be required for taking the exhaust of two Allis-Chalmers Diesel type oil engines, say, of 90 horsepower each, exhausting into one line 70 feet long?

President Doelling.—For each 90 horsepower engine I would take a 7-inch exhaust pipe and a 10-inch main for two engines. Smaller pipes could be used, but the result will be a higher back pressure, which lowers the economy of the engine. Besides, it is not desirable to exhaust into one pipe, because in case of trouble one cannot see, from the outlet, which engine is out of order. As stated in the paper, if the engines have individual exhaust pipes, you can see right away which one is in need of adjustment. However, it is done very often. We did it formerly, but now we try to avoid it.

J. N. Gruhl.—What is the consequence of fuel oil entering the high pressure air line?

President Doelling.—The result is a very heavy explosion, because the pressure is already about a thousand pounds and the temperature also is very high. To avoid that a check valve is put near the spray nozzle; that closes the valve against the oil pressure so nothing can come into the oil pipe. I know that formerly this occurred very often, especially with the Diesel engines provided with a high-pressure receiver. Very often a pressure of 2,500 to 3,000 pounds is reached and the tank lets go.

Henry Torrance.—What is the temperature of the air at a thousand pounds pressure? It would not be at the theoretical compression temperature, would it?

President Doelling.—No, it must be cooled, otherwise the lubricating oil would be burned. The temperature may be four or five hundred degrees, but it should not be higher than that. Some builders believe they should cool it down to 100 degrees, but when air of this low temperature and with 1,000 pounds pressure is admitted to the cylinder and there expanded down to 500 pounds, the air is too cool to ignite the fuel, and to overcome this defect the air must be compressed again. It is not done, except in submarines.

J. N. Gruhl.—You would then have to ignite the gas mixture by a hot bulb?

President Doelling.—Yes, but not in the Diesel engines, where ignition is caused by the heat of compression in the cylinder at about 500 pounds.

Henry Torrance.—I have just made a rough calculation to show how much ice the exhaust heat would make in the absorption system. The figures on page 26 of the paper show 2,500 B. t. u. loss in exhaust. Roughly, this is the equivalent of 2½ pounds of steam, and if we were to produce a hundred tons of refrigeration with about 200 horsepower, each producing 2½ pounds of steam, there would be total 500 pounds of steam available per hour, sufficient to make additional fifteen tons refrigeration by an absorption machine. I had expected a still greater surplus.

President Doelling.—You must not overlook the fact that the temperature of the exhaust gases is comparatively low. In our engines it is usually about 700 degrees, and as the exit temperature cannot be made lower than about 250 degrees, less than two-thirds of the waste heat is available. The gain is very small, and still, everybody who sees it, puts in a boiler right away. They never get any steam, because the heating surface of the boiler used is much too small. Again, the supply is not constant. For instance, if you use it for steam heating, and the engine is running only with half load, it may do in the winter between ten and two o'clock, but if it is a cold day and you need more heat, a boiler is required anyhow. Whether you then make a few pounds of steam more or less does not cut much of a figure. The only time there is an advantage is when you can use hot water, for then you can make use of the cooling water leaving the jacket, and heat that by means of the exhaust to 200 degrees, because in hot water heating you do not need such a high temperature as in steam heating.

Henry Torrance.—In our absorption machine generator we do not need a temperature over 230 degrees. We could just use the hot gases direct, without making steam at all. However, I agree with you that it is hardly worth while.

I have made another calculation comparing the cost of making ice with oil engines with the absorption and compression steam plants, and I find that it comes about half way between; in other words, from the operating expense in the compression steam plant we cut about thirty per cent off. That would be about thirty per cent better than the comparison figures presented, which would put the absorption plant about half way between the two other systems.

J. N. Gruhl.—What kind of lubricating oil does President Doelling advise for cylinders, what flash point?

President Doelling.—Not less than 345 degrees, as stated in the paper. The Standard Oil Company makes a special oil called "liquid base," which is the same as is used for ammonia compressors. This is a comparatively cheap oil. Some people lubricate engines with very expensive cylinder oils which cost 60 to 70 cents per gallon, the kind that is used in gas engines. But these are not adapted for oil engines: they are too thick and do not wash the piston clean; besides, the temperature in oil engines is considerably higher than in gas engines, and, therefore, the gas engine cylinder oils used generally burn, causing the piston rings to stick, and leading to trouble. Hence, the liquid base oil is cheaper and better for the purpose.

REVIEW

EXPERIMENTS ON COLD STORAGE INSULATION

EXPERIMENTS ON COLD STORAGE INSULATION. By K. Hencky, of the Physical Laboratory of the Royal Technical High School, Munich, Germany. *Zeitschrift fuer die gesamte Kaelte-Industrie*, August, September, 1915. 9 pp.

This instructive article is divided into: an introduction, reviewing the formulas generally used in insulation calculations, emphasizing the importance of correct knowledge of the surface temperatures; Part 1, describing an accurate method of measuring surface temperatures by means of thermo elements; Part 2, treats of tests made to determine (by means of thermo elements) the heat conductivity of the natural ground, as under a cellar; appendix, describing tests made to find the rate of heat emission from a warm copper plate to the air in the room.

INTRODUCTION

For calculating in a general way the heat flow through a wall a coefficient of heat transmission k is applied, as in the following equation:

$$Q = k \times S \ (t_w - t_c) \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where Q stands for the heat in B. t. u. passing through one square foot of surface S per hour per degree F. difference between the air temperature t_w on the warm side and the air temperature t_c on the cool side of a wall.

In the case of a cold-storage room, outdoor heat is first imparted to the outside of the wall, then the heat is conducted through, and finally it is transferred from the wall surface to the cool air in the room. Thus there are two heat transfers involved and one heat conduction. Accordingly, the over-all heat transmission coefficient k is composed of

$$k = 1 \div \left(\frac{1}{\alpha_1} + \frac{d}{c} + \frac{1}{\alpha_2}\right) \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

where α_1 and α_2 express the rate of heat transfer from surface to air in B. t. u. per hour per square feet per degree F. temperature difference between surface and air. d is the thickness of the wall, and c its heat conductivity.

If the wall is composed of various layers of the thicknesses d_1, d_2, d_3, etc., with corresponding conductivities c_1, c_2, c_3 etc., the equation (2) becomes

$$k = 1 \div \left(\frac{1}{\alpha_1} + \frac{d}{c} + \frac{d_1}{c_1} + \frac{d_2}{c_2} + \frac{d_3}{c_3} + \ldots + \frac{1}{\alpha_2}\right) \ldots\ldots (2a)$$

The results obtained with the foregoing equations are only acceptable if the proper values for α_1 and α_2 are known. Since these are, as a rule, not accurately known, and vary with the nature of the exposed surfaces, air velocity, etc., it is proposed to take as a basis the temperatures t_1 and t_2 of the exposed surfaces or faces of the wall. Then the following equation applies:

$$Q = \frac{c}{d} \times S \times (t_1 - t_2) \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

With this equation the heat Q_1 admitted through the four walls and roof may be calculated. Q_1 may also be made equal to $Q_3 - Q_2$, if Q_3 represents the amount of refrigeration necessary, according to test, for maintaining the room at a specified temperature, and Q_2 is the heat calculated as coming from the ground up. The refrigerating work Q_3 may be obtained by metering the quantity of brine circulated per hour through the cooling coils, and multiplying the pounds of brine with the degrees average temperature range and with its specific heat at the mean temperature. (This method requires long testing, constant conditions, and non-interference from goods in the room or the ice on the pipes.)

The heat Q_2 through the ground area S_2 may be calculated from the coefficient of heat conductivity c_1 of the ground, and the two temperatures t_b and t_t of two points separated vertically by the distance d, as follows:

$$Q_2 = \frac{c_1}{d} (t_b - t_t)\ S_2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

To use equation (3) a device is necessary for ascertaining the exact temperature of the surfaces; this is described in Part 1. To be able to use equation (4) the heat conductivity c_1 of the ground must be known. This is given in Part 2.

MEASURING SURFACE TEMPERATURES

Measuring of surface temperatures by means of mercury thermometers is declared useless, because of the impossibility to shut out the influence of the surrounding air. In cases where the surface temperature was known to be 100° C., common thermometers indicated only 80 to 85° C. A more accurate method, by means of thermo elements, adapted for metal as well as insulating surfaces, has been devised, and its theory and construction are given in detail. By tests its accuracy was found to be correct to 0.005 millivolt, corresponding to an average accuracy of 0.14° C. (¼° F.), when the instrument was applied to a surface of boiling water of accurately known temperature.

Readings of the instrument were also compared with the figures resulting from interpolating the temperature gradient obtained by inserting 6 measuring wires within a thickness of 2 inches. These temperature gradients follow straight lines, hence by extending on a diagram the lines to each face of the insulator tested, the exact temperature prevailing there is found.

Table 1. Temperature gradient through teak wood

Temperature	129.6	108.6	82.8	65.0	54.9	50.4° C.
Distance from electrically-heated surface	47.0	36.0	21.6	11.0	5.0	2.0 mm.

Surface temperature by curve, 46.1° C.; by instrument, 46.0° C. Room temperature, 15° C.

Table 2. Temperature gradient through cork board

Temperature	55.8	51.0	46.0	38.0	30.6	27.8° C.
Distance from electrically-heated surface	47.2	41.0	34.0	19.3	8.0	3.0 mm.

Surface temperature by curve, 25.9° C.; by instrument, 25.9° C. Room temperature, 15° C.

The base of the measuring thermo element is composed of a small copper disc, 1⅛ inches diameter and 1/16 inch thick. When placed on a surface, the temperature of which is desired, this disc does not alter the temperature, because it is an excellent conductor. In order to be sure that also the radiation remains as before, a thin disc of the material under test is put on the back of the copper disc. Thus it is possible to get the true temperature of any surface, only sufficient time must be allowed to let the instrument acquire the temperature of the surface of the material under the disc. With metals, 1 to 2 minutes will suffice, with teak wood it took 14 minutes, and with cork board 17 minutes. Perfect contact also is necessary.

HEAT CONDUCTIVITY OF THE SOIL

The amount of heat Q_2 entering from the ground through a basement floor of area S_f, may be calculated by the following formula:

$$Q_2 = \frac{S_f\,(t_b - t_t)\,c_1}{d_1 - d_2} \qquad (5)$$

where c_1 is the heat conductivity of the soil, and t_b and t_t the temperatures of same at the two (vertically separated) points d_1 and d_2.

Heretofore only a laboratory test had been made by H. Groeber, 1910, who ascertained for the sample of soil he tested, specific gravity 2.040 = 127.15 lbs. per cu. ft.; c_1 at 32° F., 0.43 calories (83.22 B. t. u. per 24 hours per sq. ft., 1 inch thick per deg. F.); at 68° F., 0.45 cal., or 87.1 B. t. u.; at 158° F., 0.50 cal., or 96.75 B. t. u. Since the conductivity depends greatly upon the density and moisture content, it was decided to repeat the test on a large scale. A favorably situated basement room of 146.5 sq. ft. floor area was available in a brewery in Munich, surrounded by other cellars, all kept at the freezing point. For the purpose of the test the floor was covered with cork board 0.1 m. (3 15/16 inches) thick, presented by Grünszweig & Hartmann, and the heat conductivity c_2 of this insulator tested in the laboratory of the school. Thermo elements were placed on top and bottom of the cork board on the ground for taking the temperatures t_1 and t_2. To keep moisture from the instruments waterproof cement was used as a covering.

When conditions have become constant, the heat flow from ground may be calculated as follows:

$$Q_2 = \frac{S\,(t_1 - t_2)}{d} \times c_2 \qquad (6)$$

In connection with equation (5) the unknown coefficient of heat conductivity c_1 of the soil is obtained by writing

$$c_1 = \frac{t_1 - t_2}{d} \times \frac{d_1 - d_2}{t_b - t_t} \times c_2 \qquad (7)$$

The temperature of the cork board surfaces was easily obtained, but some specially designed thermo elements had to be built for driving them into the ground to various depths including 13 feet. Full details are given of these devices. The readings were accurate to ± ¼° F. The observations extended over four months' time. The average values obtained were:

Temperature of upper face of cork board, $t_2 = 1.4°$ C.
Temperature of under face of cork board, $t_1 = t_t = 3.6°$ C.

Temperature in soil 2 meters down, $t_b = 4.5°$ C.
Substituting these values in equation 7, c_1 is obtained for the soil:

$$c_1 = \frac{3.6 - 1.4}{0.10\,m} \times \frac{2\,m}{4.5 - 3.6} \times 0.041 = 2.0 \text{ cal./m./hr./°C.}$$

where $c_2 = 0.041$ cal., the heat conductivity of the corkboard (based on 1 meter thickness). 2.0 cal. corresponds to 387 B. t. u./sq. ft./24 hrs./1 in. thick/°F., against Groeber's value of 0.44 cal., or 85 B. t. u./24 hrs., at the same mean temperature. The difference must be due to the greater water content, owing to the fact that the ground water was within 5 meters (16.4 feet) of the floor level. Thespecific gravity of the soil tested in the brewery is not stated. The necessity for ample floor insulation will be evident.

According to equations (1) and (2) the heat flow through the ceiling of a storage cellar, composed of masonry and filled with gravel ($c = 0.33$ cal.), when the temperature outside is 18° C. and inside + 1° C., equals

$$\left[1 \div \left(\frac{1}{5} + \frac{1}{0.33} + \frac{1}{5}\right)\right] \times (18° - 1°) = 4.96 \text{ calories}$$

per square meter per hour per 1 meter thickness, and through a square meter of the 5-m. thick insulated floor of, say, 10° C.,

$$\left[1 \div \left(\frac{1}{1} + \frac{5\,m}{2.0}\right)\right] \times (10° - 1°) = 3.33 \text{ calories per hour.}$$

If the cellar floor is insulated with cork board ($c = 0.04$ cal.) 0.1 meter (3 15/16 inches) thick, the heat supply from the ground is reduced to, as usually written,

$$(10° - 1°)\,\frac{1}{\dfrac{1}{5} + \dfrac{5\,m}{2} + \dfrac{0.1}{0.04}} = 1.73 \text{ calories/m.}^2\text{/hour,}$$

which means for this room a saving in refrigeration of

$$\frac{(4.96 + 3.33) - (4.96 + 1.73)}{(4.96 + 3.33)} \times 100 = 19.3 \text{ per cent,}$$

and if the ceiling is insulated the same way the total saving is 25.8 per cent. (No heat loss through walls is included, because this cellar is supposed to be enclosed by others of like temperature.) In rooms receiving heat also through the walls, the percentage of saving due to the insulation is less, because the total heat flow is greater. A diagram is added showing that while directly under a cellar floor of 32° F. the ground temperature increases downward in intervals of 1.8° F. every 2 feet, until 50° F. is reached at a depth of 20 feet (ground water temperature 50° F.), and the cellar, including its ceiling, is 20 feet high above the floor, the grade outside being level with top of ceiling, the temperature away from the cellar's walls increases from 50° F. by 1.8° F. every 4 feet upward, until, after 40 feet, when the grade line is reached, the temperature is 68° F. With this information it is possible to estimate more accurately the amount of heat entering through the walls of a basement.

MEASURING HEAT EMISSION

Heat is received and emitted by a wall or surface to the surrounding air by radiation and convection. It is customary to express this heat exchange by the following equation:

$$Q = \alpha \times S \times (t_{wall} - t_{air}) \quad \dots\dots\dots\dots\dots\dots (8)$$

in which $Q =$ the heat given off by the surface S, and α is the heat per square foot per hour per degree difference between temperature t_{wall} of the surface and temperature t_{air} of the air.

The factor α is physically not accurately defined; because the air temperature is not the same at every point, and it is uncertain as to where or how far from the surface t_{air} should be measured. For these reasons the tendency of new scientific investigations is to, if possible, eliminate α, and to convert phases of heat transfer into those of heat conduction. There are, however, cases where it is desired to know the temperature drop existing between surface and air, and therefore the following experiments were undertaken. A number of different tests will be necessary to fully cover the subject, in this article only the first series is reported:

Description of apparatus used: A heating element 60 cm. square was covered on both sides with copper wire mesh and asbestos card board. The resistance wires were capable of generating heat at the rate of about 1,000 calories (3,968 B. t. u.) per hour. This was laid on a piece of cork board of same size, but 11 cm. (4.33 inches) thick. These plates were put into a wooden box 1 meter (39.37 inches) square and 30 cm. deep, and packed on the four sides with granulated cork. The heat conductivity of the cork board had previously been accurately determined, and this, in connection with the temperature readings taken with thermo elements on both surfaces of the cork board, enabled one to determine the amount of heat lost through the insulation. The heat lost through the granulated cork on the sides is very small, at most 2 per cent of the heat generated. The greatest part of the heat (at least 95 per cent) is emitted to the air by the uninsulated surface of the warm plate.

The temperature of the warmed surface was taken by means of the surface temperature thermo element described in Part 1, while the air temperature was taken by means of ordinary thermo elements at various distances from the surface. In this series the hot plate was sheathed with ordinary rolled sheet copper, presenting a dull, oxydized surface.

Tests were made with the apparatus in vertical and horizontal positions in a small room of 812 cubic feet, 8.2 feet high, and in a large room of 8,720 cubic feet and 13 feet high, to see whether the convection currents would act differently, but the results were found to be the same. These are given in Tables Nos. 3 and 4.

TABLE 3.—HEAT EMISSION FROM A HORIZONTAL SURFACE

DATE.	Q_g	t_d	c	Q_r	Q	t_{wall}	t_{air}	$t_{wall}-t_{air}$		a		e (approx.)	
1912	Cal.	Deg. C.	Cal.	Cal.	Cal.	°C.	°C	Deg. C.	Deg. F.	Cal.	B.t.u.	cm.	Inch.
May 27	16.84	4.3	0.043	0.60	16.24	22.8	15.6	7.2	12.96	6.26	1.281	40	15.7
May 31	29.93	9.3	0.044	1.34	28.59	30.4	19.3	11.1	19.98	7.16	1.465	60	23.6
June 3	43.50	11.4	0.045	1.66	41.48	34.8	18.4	16.4	29.52	7.10	1.453	90	35.4
June 5	64.40	16.6	0.046	2.51	61.89	42.1	19.6	22.5	40.50	7.64	1.563	180	71.
June 8	93.10	21.9	0.047	3.34	89.76	51.8	21.7	30.1	54.18	8.28	1.696	240	94.4
June 10	212.00	46.5	0.051	7.69	204.31	80.0	21.0	59.0	106.20	9.62	1.969	240	94.4
June 13	320.00	58.8	0.055	10.48	309.52	103.2	23.8	79.4	142.92	10.80	2.211	240	94.4
June 15	459.00	74.8	0.060	14.55	444.45	130.0	22.6	107.4	193.32	11.50	2.354	240	94.4

TABLE 4.—HEAT EMISSION FROM A VERTICAL SURFACE.

DATE.	Q_r	t_d	c	Q_r	Q	t_{wall}	t_{air}	$t_{wall}-t_{air}$		a		e (approx.)	
1912	Cal.	Deg. C.	Cal.	Cal.	Cal.	°C.	°C.	Deg. C.	Deg. F.	Cal.	B.t.u.	cm.	Inch
June 20	5.92	2.0	0.044	0.29	5.63	23.8	20.4	3.4	6.12	4.60	0.942	40	15.7
June 22	16.80	5.1	0.045	0.74	16.06	29.2	21.2	8.0	14.4	5.58	1.142	70	27.5
June 24	31.00	8.6	0.045	1.25	29.75	35.8	22.4	13.4	24.12	6.16	1.261	90	35.4
June 28	162.00	38.3	0.051	6.34	155.66	76.5	20.7	55.8	100.44	7.76	1.588	150	59.0
Jluy 3	217.60	49.9	0.053	8.53	209.07	89.3	21.0	68.3	122.94	8.50	1.740	150	59.0
July 5	436.00	81.2	0.060	19.40	416.60	141.3	20.6	120.7	217.26	9.74	1.993	150	59.0

Notation for Tables Nos. 3 and 4:

Q_g = total heat generated in hot plate in calories per hour.
t_d = temperature difference between the two faces of the cork board in degrees C.
c = the internal heat conductivity of the cork board at the mean temperature used in test, expressed in calories per m^2 per hour per °C. per m thickness. (One of these units = 193.527 B. t. u. per sq. ft. per 24 hours per °F. 1 inch thick; $c = 0.044$ cal, corresponds to 8.52 B. t. u./sq. ft. per 24 hours 1° F. 1 inch thick.)
Q_r = heat leakage through the cork board, determined from t_d and c, in calories per hour.
$Q = Q_g - Q_r$ = the heat emitted from the warm surface to the air in calories per hour.
t_{wall} = surface temperature of the sheet copper.
t_{air} = air temperature in °C., measured at distance e from the warm surface.

$$\alpha = \frac{Q}{S \times (t_{wall} - t_{air})}$$ = the coefficient of heat emission in calories per hour per m^2 per °C. difference, and in B. t. u. per hour per square foot per degree F. difference.

At the distance e from the surface the convection currents equalize, the temperature there being identical with the room air temperature. The figures indicate that distance e increases as the temperature of the surface is increased. When the surface is held in a vertical position, distance e is perceptibly smaller than it is with the horizontal position. Both of these features are explained by the relative intensity of the convection currents and its direction with respect to the warm surface.

Fig. 1 shows graphically the variation in the emission factor α, with increasing temperature differences $t_{wall} - t_{air}$ with horizontal and vertical plates.

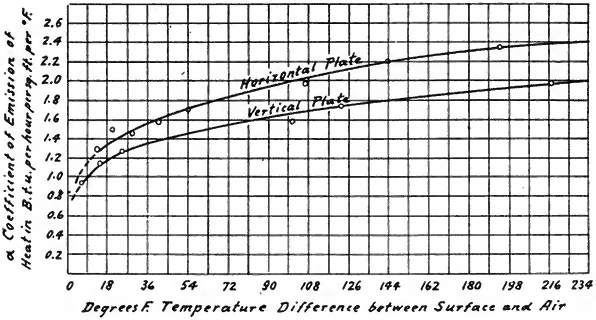

Fig. 1.—Rate of heat emission from a flat sheet, copper surface

The emission from a horizontal surface was found to be more than for a vertical one, due to the difference in convection, as stated, which is as yet not taken into account in the formula. A more exact explanation can be given only after further tests have been made regarding velocity and behavior of the convection currents.

Therefore, the constants given in the tables represent average values only, applying to the conditions of the test method. Since, however, with enlarged surfaces the convection currents may be expected to act in the same way, it is permissible to use the values given for *a* for surfaces of other dimensions with a fair degree of accuracy.

DEVELOPMENT OF REFRIGERATION IN THE UNITED STATES

Development of Refrigeration in the United States. By J. F. Nickerson, Chicago, Ill. Paper read before the International Engineering Congress, San Francisco, Cal., September, 1915.

Within the time of the present generation practically the entire industry of refrigeration has grown from a crude or experimental stage to its present magnitude, serving as an outlet for the best efforts and endeavors of hundreds of thousands of people, contributing to the advancement of the industrial arts, conserving millions of dollars' worth of food products annually, equalizing extreme prices for seasonable products, sharing in the advance that medical science has made, increasing the supply of perishable food stuffs by greatly extending the market for same, thus stimulating production, and furnishing our tables with fruit and other delicacies that the previous generation considered luxuries, or were unable to secure. Withal it has been a leading factor in making the fullness of present-day life.

A brief historical review of the evolution of mechanical refrigeration in the United States from the theoretical and experimental to the practical stage may be of interest. The production of ice seems to have been the chief incentive to the work of the early inventors and engineers, but the requirements of the brewing, meat-packing and other industrial establishments, and the application of refrigeration for the preservation of perishable food products, are largely responsible for the remarkable development of mechanical refrigeration during the last thirty years.

EARLY ICE MACHINES

The constant effort of man has been to substitute for the cold natural atmosphere, which abstracts the latent heat from water in a slow and irregular fashion, a more powerful and quickly applied agent under his absolute control. As early as the middle of the eighteenth century, scientists were experimenting upon "nature's abhorrence of a vacuum" to produce evaporation and refrigeration. It was by the application of this principle that Dr. Wm. Cullen invented a machine in 1755 which was the first to produce ice by purely mechanical means. He reduced the atmospheric pressure upon water in a closed vessel with an air pump, the evaporation from the surface of the water being so increased as to produce intense refrigeration and ice. This was the pioneer ice machine, not only of the vacuum type, but of any kind.

A few years after the invention of the vacuum ice machine the affinity that sulphuric acid has for water had been discovered, and in 1810 Leslie made ice through this agency. In the meantime the Royal Institute of England had been founded (1799) and in 1823 Sir Humphrey Davy and Michael Faraday (chiefly the latter) had demonstrated that gases could be liquefied by pressure, with continuous cooling to carry away the heat developed by work done on the gas. Volatile freezing mixtures were also used in a vacuum.

Jacob Perkins, an engineer and inventor, born in Newburyport, Mass., July 9, 1776, is generally accorded the credit for inventing the first machine, which was the forerunner of the modern compression apparatus, capable of producing ice in commercial quantities. His patent was obtained in England, August, 1834 (No. 6662)—ether being the refrigerant employed. The evaporator containing the liquefied ether enclosed a system of pipes through which brine circulated, whose temperature was thus lowered to 5 deg. F. The brine then passed into a long receptacle containing boxes filled with water, and having frozen their contents, was pumped back to be again subjected to the refrigerating effect of the expanding ether. Thus the cycle was completed and the principles established upon which most modern refrigerating machines are founded. The brine, or indirect system, was thus foreshadowed. The Perkins apparatus included the four principal features still in use in all compression systems of to-day, viz., the compressor, the condenser, the expansion or regulating valve, and the evaporator. It was also constructed according to the can system of ice-making.

Prof. A. C. Twining, of New Haven, Conn., is the next figure to issue prominently into view as a practical inventor of ice-making machinery, although Dr. John Gorrie, of Apalachicola, Fla., had been experimenting for a number of years previously with the cold-air type class of apparatus. After having worked since 1848 with sulphuric ether as a refrigerant, in July, 1850, Prof. Twining took out a patent in England, and in November, 1853, the U. S. Patent Office issued papers to him (No. 10221) for the same mechanism. Prof. Twining discovered, among other things, that water frozen at a temperature slightly below the freezing point would produce transparent ice, except the core, while, if frozen at lower temperature, the ice cake would be opaque throughout. He is said to have had a machine in operation in Cleveland, Ohio, about 1855, which produced ice at the rate of 1,600 pounds in twenty-four hours. Prof. Twining's machine was considered such a practical success that he was requested to prepare estimates for a plant in New Orleans, capable of producing eighty tons daily. This he did, estimating "the total cost of $160,000, with machinery complete, buildings and grounds; and the daily expenses, maintenance, fuel, wages, oil, repairs to machinery and building, ether and all contingencies, at $120 per day, to which was added $26.30 per day for interest at 6 per cent on the investment, bringing the total daily cost for eighty tons up to $146.30, or $1.83 per ton." Although these figures are considered remarkable to this day for their accuracy, the scheme failed for lack of support.

To Dr. John Gorrie was issued the first patent in America for the manufacture of ice by mechanical operation. Dr. Gorrie, in the line of his professional work, commenced his experiments with cold-air machines as early as 1845, his design being to use them in the cooling of sick rooms and in the treatment of fevers and other diseases in which "cold" could be used as an alleviating agent. His machine had a compression cylinder, the air passing from it into a cooling chamber immersed in water. The patent (No. 8080) was issued on May 6, 1851, the letters patent to run from August 22, 1850. It is said that he actually made a small quantity of ice with a model machine at an Apalachicola hotel about this time; however that may be, his was undoubtedly the forerunner of the compressed air machine, later improved upon. Although Dr. Gorrie was a resident of Apalachicola, Fla., he was stated, in the letters patent, to be of New Orleans, the explanation being that those who loaned him money with which to conduct his experiments required this concession from him. The inventor and physician died in the former city June 16, 1855, virtually abandoning his practice during his latter days in order to develop his ice machine, although highly esteemed for his professional talents.

AMMONIA ABSORPTION MACHINE

It was Ferdinand P. E. Carré, of France, who in 1858-1860 placed upon the market a machine which gave birth to the ammonia absorption machine system of to-day. The original machine was a very crude affair, consisting merely of two vessels—one surrounded by cold water, the other containing the ammonia and water. The original patent in the United States (No. 30201) was issued October 2, 1860, the reissue being dated February 18, 1873.

In 1863, during the Civil War, Mr. Bujac, of New Orleans, La., shipped through the blockade and erected in Augusta, Ga., a small Carré machine having a supposed capacity of 500 pounds of ice per day. The machine, however, was not a success, due mainly to improper handling by those in charge. This machine was removed to Gretna, La., in 1866, to be used for exhiition and experimental purposes.

The firm of Bujac and Girarde, of New Orleans, had imported three other Carré machines from France, purchasing the patent rights for the United States. These machines were erected in New Orleans, but were not successful.

In the fall of 1865 the firm of Mepes, Holden, Montgomery & Co. purchased one of three Carré machines imported from France and shipped it to San Antonio, Texas. It was erected and put to work at that place under the supervision of the senior member of the firm, D. L. Holden. Although this machine was erected in accordance with the plans and specifications of Mr. Carré, after running it one season many changes had to be made, the principal one (and the first time this was ever done) being the placing of a steam coil in the still of the machine for the purpose of generating ammonia gas. At this time resort had to be made to distilling the water on account of the lime and magnesia in it, when—very much to the surprise of those operating the apparatus—the ice came out transparent, which was the first transparent ice ever made by any ice machine on a commercial scale.

In 1868 the Louisiana Ice Manufacturing Company erected six 10-ton Carré absorption ice-making machines, which were constructed at Gretna, La., opposite New Orleans, by Sylvester Bennett, from plans furnished by Mr. Carré. The retorts or generators were made of heavy, double-riveted boiler iron and were heated by direct fire. The freezing tanks were made in the shape of a cross, each tank being about 7 x 6 x 4 feet deep. The cans holding the water to be frozen were 2½x10x30 inches deep, and when frozen weighed twenty-five pounds each. Five blocks were placed one upon the other, making a block of 100 pounds. This ice was not clear nor very solid, and was made of filtered river water, its color resembling stained alabaster. The public did not take to it kindly. The froth or foam arising from the surface of the water containing pieces of this melted ice gave rise to several objections, the most serious and common being that the ice contained ammonia and was unwholesome. The objection, although not real, caused the Louisiana Ice Company to employ a chemist to find, if possible, a remedy, selecting Dr. Joseph Albrecht of the United States Mint (New Orleans), his advice being to boil the water before it was put into the cans, so as to drive off the air and gases it contained. The experiments in this direction were successful; susequent experiments being made which led to the use of condensed steam instead of hot water.

In April, 1867, Prof. P. H. Van der Weyde, of Philadelphia, Pa., patented the use of naphtha, gasoline, petroleum-ether and other condensed petroleum gases as refrigerants and in February, 1869, the United States issued his letters patent (No. 87084) for his ice machine. His apparatus was of the vacuum type, consisting chiefly of an exhaust and force pump and cooling oil, and was arranged to produce both round and square pieces of ice. His machines were not utilized to any great extent.

Shortly after the installation of the Carré machine in New Orleans (1868), mentioned in the foregoing, Charles Tellier, of France, took out patents on his compression apparatus, whose refrigerating agent was methylic ether, and which was designed to make ice and refrigerate air and liquids. The date of his letters patent in the United States was January 5, 1869, and in that year George Merz, a prominent brewer of New Orleans, imported a small machine from the works of the inventor and erected it in his plant, with the object of producing cold, dry air and of making ale and lager beer without ice. As it was designed for ether, but ammonia was used, it did not work well and was finally discarded, although a commencement had at length been made in America of artificial refrigeration in breweries.

In September, 1869, April, 1870, and at various later dates Mr. Holden obtained patents on his refrigerating apparatus. His first patent was No. 95347, dated September 26, 1869. The machine (known as the Regealed), which was the culmination of Mr. Holden's thirty-five years' experience in business, forms the ice on the outer surface of a revolving cylinder from which it is removed by a scraper, the resulting flakes or scales being afterward compressed into cakes. The apparatus is designed to use either the compression or absorption system.

John M. Beath was another early and practical American inventor of ice-making machinery. In connection with Samuel B. Martin he constructed an ice plant during 1869 at Los Angeles, Cal., and in 1870 and 1871 at San Francisco and Portland, Ore., respectively. These plants were operated by compression machines and the ice was made in the first two by the plate method and in the last by sprinkling the water to be frozen upon freezing or congealing surfaces. The first patent was granted to Beath and Martin as joint inventors May 28, 1872 (Martin having furnished the capital), and was reissued on July 16, 1872, to make it stronger. The patent specification described the first flooded system in which the flow of refrigerant to the evaporating coils was governed by the quantity of liquid in the evaporator itself.

In 1872 Mr. Beath constructed an ice plant in Chattanooga, Tenn., using an absorption refrigerating machine. This plant operated successfully the following winter. The ice was frozen as in the Portland, Ore., plant, on vertical pipes over which water was sprayed by rotating sprinklers. The ice was allowed to accumulate until the space between the pipes was filled and, although considerable labor was experienced in cutting this ice out. also with no small percentage of waste, this factory caused a permanent reduction in the price of ice in Chattanooga of over fifty per cent. Beginning in January, 1874, Beath applied for three patents covering the ice-making machinery in this plant. The atmospheric coil absorber used was a main feature. These patents were all issued in 1875. The success of the Chattanooga factory was such that Beath organized in 1874 the North American Ice Company and erected plants in Galveston, New Orleans and Atlanta in 1875. In the interval between the erection of the Chattanooga and the other southern plants, Beath returned to San Francisco and improved and enlarged the plant in that city.

Patent office records show that besides the patents mentioned above, one was granted April 13, 1875, to Beath and Martin as joint inventors and to Martin as sole inventor on May 5, 1874, and April 20, 1875.

In the early seventies the so-called cold-air or compressed-air machines monopolized to a great extent the attention of both inventors and the commercial world of refrigeration. Franz Windhausen, of Brunswick, Germany, was the foremost exponent in this field, and his machines were soon installed in German and American breweries. The original letters patent were issued in the United States on March 22, 1870, and are described as relating "to that class of freezing and ice-making machines in which atmospheric air is compressed, then passed through a cooler and afterward expanded to remove the heat (or, in other words, to produce cold), but which machines have hitherto been limited in their cold-producing properties by the degree of expansion to which the air is subjected and by the temperature of the cooling water employed." Windhausen patented his vacuum machine in Germany in 1877, and subsequently in the United States. In 1878 he invented his compound vacuum pump for producing ice on a large scale, and greatly improved it in 1881, so that his machine would produce fifteen tons of ice in twenty-four hours—which was considered remarkable for that time.

AMMONIA COMPRESSION MACHINES

A few years after the Windhausen cold-air machines were placed on the market another German inventor came into the historic foreground, Prof. C. P. G. Linde, of Munich, Bavaria, who introduced his ammonia refrigrating machine in 1873-1875. He is now generally considered one of the foremost early exponents of the compression system. The first attempt at putting the machines in operation was made in 1873, and in 1875 was erected a compressor, No. IV, for sweet-water cooling in the Spaten Brewery, Munich. Prof. Linde patented his machine in the United States in 1880. The first Linde machine in the United States was erected in the brewery of the Wacker & Birk Brewing & Malting Company, Chicago, in 1880, by Fred W. Wolf.

Capt. David Smith, the inventor of the plate system of ice-making, was a native of Cape Cod, town of Chatham, Mass., born in 1829. He built his first machine at San Francisco in 1872, and, according to the statement of his nephew, Curtis M. Smith of Washington, D. C., it was a one-ton can machine. This he took to Honolulu, and although he made some ice he had trouble in confining the ammonia; whereupon he put his ammonia pipes inside of a large box. He filed an application for a patent based on this idea in November, 1875, and letters were granted him (No. 173357) on February 8, 1876. Captain Smith was the first to freeze water on the outside of a congealing box with a coil of pipe immersed in brine, or in other words to use the plate system of making ice.

Raoul P. Pictet, professor of physics in the University at Geneva, Switzerland, had been experimenting for several years with the liquefaction of gases and refrigerating apparatus, prior to taking out his first patent in the United States in February, 1877. Pictet found that pure anhydrous sulphur dioxide may be successfully employed in an apparatus suitable for effecting therein its continuous alternate vaporization and condensation, so as to successfully manufacture ice and refrigerate air on a practical scale without injury to the apparatus, or the formation of ice within the vacuum or condensing pump.

One of the earliest inventors and builders of refrigerating machinery in the United States was Thomas L. Rankin, a native of Ohio, whose earliest experiments were conducted in the South. His machines are of the absorption type and are used in not a few of the breweries of the country. From 1868 to 1884 no less than twenty-four patents were issued to him. The application of his apparatus included its use for beer coolers, cold storage houses and refrigerator cars, for the manufacture of ice, for skating rinks, the refrigeration of breweries and packing houses, vessels carrying meats, etc. The first patent on an ice machine issued to Mr. Rankin was No. 175,498, dated March 28, 1876.

David Boyle, a Scotchman and a close associate of D. L. Holden, Thomas L. Rankin and other pioneers, made a remarkable practical success of his inventions. From 1875 to 1878 Mr. Boyle was associated with W. B. Bushnell, of Quincy, Ill., building ice-making and refrigerating machinery in Chicago, much of which was shipped to Texas. In 1878 the Boyle Ice Machine Company was incorporated in Chicago, and that company built a large number of ice-making and refrigerating machines.

In the early part of 1876, F. M. McMillan and Silas Merchant founded the firm of F. M. McMillan & Co., in Cleveland, Ohio, for the purpose of manufacturing and selling ice-making machines. They employed John Enright as their supervising engineer. The machine which they proposed to build was one designed by either Merchant or McMillan, and had a single-acting compressor. No patent appears to have been taken out for this apparatus, although previous to the organization of the Arctic Company it was known as the Silas Merchant machine. In 1877, Mr. Enright designed and built a machine having a vertical double-acting compressor. The patent for this compressor was No. 202,641. With the making of their machines in 1876, or at least soon after, F. M. McMillan & Co. commenced the manufacture of anhydrous ammonia, although a patent for the drying of the gas was not issued to Mr. McMillan until March, 1879.

Charles J. Ball installed his first absorption ice-making machine at Sherman, Texas, in 1878, it having been constructed by him in St. Louis in the fall of that year. It was a modified Carré machine, the ice tank being only eight feet square, and the cans made of galvanized iron and copper. The ammonia pumps, water pumps, boiler feed pumps, brine pump, etc., were driven by belt from shafting run by a riding cut-off slide valve engine. The machine made about five tons of ice daily with one ton of coal, the actual cost of installing same being $12,000.

In 1879 the first De La Vergne refrigerating machine was placed in the Herman Brewery at No. 221 West Eighteenth street, New York City, the inventors of the original apparatus being John C. De La Vergne and William M. Mixer. The material improvement claimed for their patent over other refrigerating machinery based upon the compression and expansion of ammonia, was the effective means adopted for sealing the valves and joints and lubricating the internal mechanism of the compression pump, by the forced circulation of oil both ahead and behind the compressor piston as stated in the patent specifications.

In March, 1879, Larkin & Scheffer, of St. Louis, started an experimental compressor to liquefy and ship anhydrous ammonia in suitable containers. It was in 1880, the year succeeding, that the Mallinckrodt Chemical Works of that city founded a plant for the manufacture of anhydrous ammonia. These were the forerunners of similar plants afterward established at various points in the United States.

The decade from 1880 to 1890 marked a very considerable activity in the production of new apparatus. Among the American inventors of refrigerating machinery to whom patents were issued during this period may be mentioned the following: Thomas L. Rankin, Cassius C. Palmer, Thomas Cook, Charles G. Mayer, George W. Stevens, Richard Thoens, Francis V. De Coppet, Victor H. Becker, Fred W. Wolf, John Ring, A. F. Ballantine and H. D. Stratton.

A QUARTER CENTURY OF PROGRESS

It is just twenty-five years since there occurred one of those fortuitous incidents which so often mark radical changes in the development of an industry. Through the art of refrigeration considerable good work had been done in breweries and packing houses in the manufacture of ice and the cold storage of foods in a small way. But the methods used were singularly individualistic; each manufacturer had his own type of apparatus, even to the smallest detail: there was to a very large extent no real refrigerating engineering. This and that was tried and if it proved effective its use was continued. Results were sought with but little regard as to how those results were obtained. Abroad there was some advance being made in the sciences applicable to the art.

During those dark ages, as they might be called when speaking of refrigerating engineering, the individual received little, if any, assistance from the work done by others, practically no one published the results of what experiments they had made, and in many instances many men were spending their time and energy on the same problem. That there were men at that time who will always rank high in the list of refrigerating engineers there can be no doubt, and there is no desire to belittle the wonderful advance that was made, but had there been the spirit of co-operation, refrigerating engineering would have come into its own long before it did.

There were practically no standards: there were few, if any, attempts to treat the problems in a scientific manner, insulation in many cases was lost sight of, and losses were fought by expenditure of energy that might have been conserved.

There had also come about a diminution in demand for refrigerating machinery; many manufacturing companies saw no great future to the industry, and took no steps to build up their plants, contenting themselves with making as much as they could from the business offering, believing that the volume would diminish rather than increase.

Such was the condition when manufacturers were called upon to meet what was, and will probably remain, the greatest shortage in the natural ice crop; from the Mississippi to the Atlantic Coast the cry went up for ice. While in a measure this circumstance was disastrous, for some undertook to supply in a limited time an amount of equipment that was so far beyond their capacity that they could not make the deliveries guaranteed, yet it awakened the public to the use of mechanical refrigeration as nothing else could. And thus the need for ice was again, as at the beginning, the chief incentive for the efforts put forth by the inventor and the manufacturer of refrigerating apparatus. The impetus given at that time was unquestionably the exciting cause of the wonderful development which the past twenty-five years have to record. It is well worth our while to pause and consider this quarter century of progress.

In the development of the refrigerating engineer himself is to be found the keynote of the success achieved. As soon as it was apparent that in refrigerating there was something that could supply a long-felt need, men with scientific and mechanical training began to consider the problems. Some there were who were so narrow that they could not see that by associating themselves with those who had ideas much could be accomplished. These men pass into oblivion in spite of the good work they had done. On the other hand, there were those who were quick to see that a future did exist, that others could teach them something that could be applied to their apparatus; they left that old individualists' path they had been following and struck out with others along new trails.

An important factor in this new order of affairs was the establishment of the first trade journal in the world devoted exclusively to the ice and refrigeration industries.

In every industry the trade literature has had an important bearing on the development of the industry. This was particularly true with a new and chiefly experimental industry as was the refrigerating industry in 1890. Every manufacturer, designer and builder of refrigerating plants was busy with his own problems, and, having solved same, imagined that he was the only one who knew the secret and the world was his. There was no medium of interchange of thought and experience, no journal interested in disseminating news. awakening interest and promoting the industry in general. The first to step into this sphere of activity was *Ice and Refrigeration*, conceived in 1890 and born July 1, 1891, when the initial number appeared. There were no books at that time available for the practical study of the science and application of refrigeration and the first book on this subject prepared in America was "The Compend of Mechanical Refrigeration," now in its ninth edition. This work was followed by other books until at the present time there is an extensive library on the subject. The success of the pioneer trade journal in the refrigerating field brought into being a few years later other similar journals in various European countries, and some in this country, two of which are still in existence. These publications, by arousing interest and pointing out the vast field open for the application of refrigeration, stimulated scientific investigations and, by offering a medium through which the results obtained by the most expert in any part of the world were almost at once made known to all others working in the same domain, were able to exert a vast influence in the development of the industry.

REFRIGERATING SOCIETIES ORGANIZED

In 1904, there was organized The American Society of Refrigerating Engineers, in which, as it exists to-day, the engineer finds that co-operation and assistance needed to advance the art. Dead work has to a large extent been eliminated—one now begins where the other has left off, data is considered, criticised and the help that results makes possible the rapid development of the idea.

As soon as the problems were attacked with scientific methods, it was found that much of our basic data was wanting or, if it existed, was wrong—something must be done. Much help was had from certain technical institutions and certain manufacturers did yeoman service toward getting at the facts. An elaborate testing plant was erected at York, Pa., and most careful tests were made, the results of which were freely given out. But the engineers and the industry hungered for more and for that which would be authoritative. This required, besides money, the service of men trained in scientific investigation and research. The engineers themselves were not strong enough to secure this, although they had been urging it. Fortunately there had grown up in this country an association of men interested primarily in the commercial side of the industry, but who recognized the value that would accrue to all if these matters could be settled. This organization, known as The American Association of Refrigeration, put its shoulder to the wheel and things began to move. There has been built up what is known as the International Association of Refrigeration composed of those throughout the world who are interested in the problems of refrigeration and which to-day is the most powerful instrument for the development of the art that there is. The Third International Congress of Refrigeration, in conjunction with the great Refrigeration Exposition held in Chicago, September 17 to 24, 1913, under the auspices of the International Association, brought together a large number of the world's best thinkers and practical experts in refrigeration, including official government delegates from over thirty countries, resulting in the interchange of ideas, the statement of problems and the attempts at their solution, and was of inestimable value in stimulating desire for further research and experiment.

The influence that the American Association was able to bring upon our government resulted in getting its co-operation in solving many problems that had arisen. And in the United States Bureau of Standards was found the instrument that could undertake a work not possible by private enterprise. When this bureau has determined something it speaks with authority. It has already settled that long-disputed question as to the latent heat of ice, and is now at work on other problems whose solutions engineers and the industry await with keen interest. With organizations striving to assist in the development, and a body of refrigerating engineers devoting their lives to the efficient application of the art to the needs that arise and who are also pointing out where refrigeration can serve mankind, all of which has come about in largest measure during these later years, we can naturally look for a wonderful advance, and in this we are not disappointed. But as we stand to-day and look about us from the vantage point that has been gained, it can be seen that the future holds out promises for greater achievements and he who writes of the next twenty-five years will have much to tell; but then it will be seen even more plainly than now how the past twenty-five years laid the broad foundations for future success.

MODERN IMPROVEMENTS

To recount all the improvements made would be an endless task, and yet it is well to call to mind some of the more important ones, and this will be sufficient to repudiate the statement which has been made that the refrigerating engineer has not made the progress that can be shown in some other lines of engineering.

In machine design the layman may perhaps not notice the changes which have taken place, but the scientific consideration of the problems of compressors has enabled the manufacturers to make their compressors more efficient without much change in appearance. There has, however, been one change in compressors which is apparent to the eye and that is where formerly the water jacket enclosed the entire cylinder, it is now confined to the end where the heat is generated; this is logical, for the water is warmer than the gas returning from the evaporator and if brought into contact with that part of the compressor which is cooled by the incoming gas it will have a tendency to impart heat to the gas, thus expanding it, which is just what it is desired to avoid. On the other hand, if the water can be brought into more intimate contact with that portion of the compressor which is heated in the process of compression it will be able to accomplish more good. Therefore, one now finds the heads well water jacketed, while the suction ports are often insulated to prevent any heat absorption.

It is now well recognized that the matter of filling the compressor with as much weight of ammonia as possible is most important. Compressors are in service using what is termed a "multiple effect" whereby these compressors are filled with the returning ammonia at a higher pressure than exists in the main evaporator, which is accomplished by opening a connection to the compressor after it has filled with the gas from the main evaporator, with a line from some source where the pressure is high. The result is that this higher pressure does part of the work of compressing the cylinder full of gas. This higher pressure may come from another evaporator service maintained at higher temperature; or from a special evaporator where the ammonia liquid is cooled. Then, again, rotary blowers have been installed on the line leading from the evaporator whereby the gas is raised in pressure with a less expenditure of energy than would be needed in the main compressor, and the main compressor is increased in capacity by being filled with gas at a higher pressure.

Much thought has been given to getting the compressor to fill to as near as possible the pressure in the suction line, and this has been accomplished by overcoming the resistance offered by the suction valve. To-day there are being tried various ways to further increase the efficiency. As low temperatures are demanded some way must be found to compensate for the loss in compressor efficiency due to low temperature work. One of the ways is by the use of the multiple effect compressors.

The first patent on a multiple effect compressor was issued to Gardner T. Voorhees of Boston, Mass., July 4, 1905, having been filed July 30, 1903. Besides the customary discharge and suction ports for a compressor, Voorhees proposed an additional suction port with a valve governing it, opening into the cylinder from a source of higher pressure gas. The operation consisted first of filling the cylinder with low-pressure gas and then allowing the higher pressure gas to enter, partly compressing the gas already in the cylinder, the piston then compressing the mixture.

Other patents have since been granted to Mr. Voorhees covering a different application of multiple effect principle to compressors and also a patent on a multiple effect receiver.

The use of internal combustion engines and electric motors is compelling the manufacturers of compressors to construct these so that they may operate successfully at speeds commensurate with such engines and motors, and this is being accomplished more rapidly than some are aware. A number of such plants are now in successful operation, producing refrigeration at extremely low cost. In the case of ice factories making raw-water ice, the elimination of the boiler plant and distilling apparatus has materially reduced the items of investment and depreciation.

The ammonia absorption machines, operating on the principle that water absorbs ammonia gas at a low temperature and pressure and realeases this gas at high temperature and pressure, have been greatly developed during the last twenty-five years and brought to a very high state of efficiency and simplicity, and are now in frequent use, and bid fair to become more and more a factor in plant economy, particularly for low-temperature work. Especially is this true of the exhaust steam absorption machines, which are making for greater plant efficiency, as they can be made to develop a large amount of valuable refrigeration from the exhaust that has heretofore been wasted. Truly no greater advance can be made in any industry than to find a use for that which is being wasted.

While ammonia machines have been principally mentioned there have been as marked improvements in machinery using other than ammonia as the refrigerant. This is particularly true of the carbon dioxide machines, the efficiency of which has been greatly increased and a large number are now in successful operation. They are claimed to be specially suitable for hotels, restaurants, on shipboard, etc. Some small SO_2 machines have also been introduced and also an ethyl chloride machine having a rotary compressor. As the field of refrigeration widens, the question often arises, what refrigerant is best for a given purpose. No longer does one buy refrigerating equipment simply because it can produce refrigeration, but the purchaser now asks for the type of equipment that will be the best and most efficient for the purpose for which it is to be used.

The ammonia condenser for many years was considered as efficient as was needed; but here too the desire for more effective results has wrought a revolution. Scientific attention has been given to their design and construction and it has been found that condensers can be made which will discharge the ammonia at a temperature within a very few degrees of the water used for cooling, and at the same time will keep the condenser pressure lower, both these results making for greater efficiency and being accomplished with less than half the amount of surface formerly required. Among the improvements which are radical departures from the old type are the double-pipe ammonia condensers, which have come into general use, but the latest development along this line is what is termed the flooded condenser, in which the hot ammonia discharged from the compressor comes in contact with liquid ammonia and is quickly condensed, resulting in a heating of the liquid ammonia. This heated liquid is then cooled by water in the usual way, but more easily, for here there is liquid on either side of the walls of the condensers, which makes possible a much more rapid transfer of heat for a given surface than would be possible where the ammonia is a gas. Many inventors and engineers have contributed to the success of the newer forms of condensers, notably Westerlin and Campbell with the double-pipe apparatus, and Louis Block and Thomas Shipley with the flooded type of condenser.

One of the important advances which have been made is in what is termed the flooded gravity system of feeding the ammonia. H. J. Krebs, of Wilmington, Del., was granted patent No. 436,003 September 9, 1890, which seems to cover the present flooded system. Krebs' patent involved the use of an accumulator placed above the evaporating coils. The level of the refrigerant in this accumulator was maintained automatically and from this, with the aid of gravity or mechanical means, the coils were "flooded." The discharge from the evaporating coils entered the top of the accumulator, where any entrained, unevaporated refrigerant was separated from the vaporized part before the latter was allowed to return to the compressor or absorber. Increased capacity of the evaporating coils was claimed to be secured by this system.

By the use of this accumulator with the addition of a valve on the suction line, Krebs was able to maintain a higher pressure in the evaporating coils than existed in the suction line. This feature enabled the operation of different coils at different pressures and temperatures, while all were connected to the same machine. This system, although old, has only lately been revived and, while proving very efficient under certain conditions, requires still considerable scientific and experimental work to be done upon it.

Primarily this method is based on the fact that the colder the gas, the greater its density, so that after the ammonia has been converted into a gas in the evaporator it is desirable to get it to the compressor as near the temperature at which it was converted as possible. The greater weight of ammonia thus handled by the compressor for a given displacement more than offsetting the small gain that might be had from superheating the gas on its way to the compressor. It is possible with this method to keep the evaporator filled with liquid ammonia, allowing the more rapid flow of heat through the walls of the evaporator. This is especially true of ice or brine tanks where the liquid to be cooled is on one side and the boiling liquid ammonia on the other.

The shell type of brine coolers were the first to use this method, later its use with pipe coils as evaporator has become general. Very naturally there had to be developed along with this the various types of accumulators or separating tanks, so as to present the liquid returning to the compressor.

Combined in these accumulators is the principle of returning direct to the compressor that portion of the liquid ammonia which is evaporated in cooling the liquid from the temperature of the condenser to that of the evaporator.

In the shell type of brine coolers this is easily effected, but in coil tanks there is required some way of having only that ammonia liquid which has been cooled to the evaporating temperature pass to the coils, and here is where the varied type of accumulators, as they are called, come in. There are some, however, that consider it desirable to allow this gas generated in the cooling of the liquid to pass through the coils on the theory that the liquid is thereby agitated in the coils and the net result is a gain.

In connection with these methods of handling the ammonia the greater need has come for the use of thermometers, in order to determine first that the liquid ammonia is cooled as much as possible with the water available for cooling; secondly, to know that the gas coming from the evaporators is as little superheated as possible. The use of thermometers in refrigerating plants is on the increase; and just so, for the whole process is one of heat transmission, which can only be properly considered when temperatures are known. It is axiomatic in all work in refrigeration that as much of the work as possible be done by natural means before applying mechanical. This is well evidenced in the cooling of the distilled water to as near the temperature of the water available for cooling as possible, which means much in the way of saving of mechanical refrigeration.

IMPROVED METHODS OF ICE MAKING

The demand for the manufacture of ice where forms of power other than steam engines are available, brought about by the decreased cost of power through the use of internal combustion engines or electric motors, used in connection with ammonia or carbonic acid compression machines, has introduced a great variety of methods for the manufacture of ice. The plate system, one of the earliest methods of making ice, has been greatly improved and made more efficient, and other systems, such as "center freeze," Holden's process of compressing into block ice shavings cut from revolving drums on which ice has been frozen, and the Patten vacuum system, making opaque ice, have been introduced and experimented with on a large scale.

The greatest development, however, has been in the manufacture of first-class merchantable ice in cans from raw or undistilled water. This is now an accomplished fact, and while there are a great variety of methods the general characteristic is that the water in process of freezing is kept in agitation by means of air discharged into it, or kept in motion by mechanical agitation. This has made it necessary to solve many problems connected with the water itself. Certain impurities affect the ice made in this way to such an extent that these impurities, usually in the form of solubles, must first be removed.

A patent was issued August 6, 1901, to Edgar J. Ullrich of Colorado Springs, Col., for an attachment to an ice can for the purpose of supplying compressed air or gas at the base of the can to produce circulation in the water in the can during the process of freezing. This appears to be the first patent issued covering this method. Other patents have since been issued, covering various systems or parts of apparatus for the production of crystal ice in cans from raw water, to Simon, Berryman, Parsons, Pownall, Beale, Shipley, Jewell, Fisher, Pharo, Maginnis and others.

Greater efficiency has been attained by the distilled water plants also through more economical steam engines and the use of multiple effect evaporators. To-day it is possible to produce a uniform quality of product at a reduced cost.

Large brine tanks with coils therein for refrigeration uses other than ice making have been largely superseded by the use of brine coolers either of the shell type or the double-pipe type, and in many instances ice tanks have been and are being constructed without coils wherein the brine is cooled by such brine coolers frequently placed in the ice-making tank.

Street pipe-line systems of refrigeration, which at one time bid fair to become important factors in the production and distribution of refrigeration, seem to have encountered so many difficulties that little has been done in this direction during recent years. At the present time there are only two large installations of this kind—one at St. Louis, Mo., operating on the ammonia direct expansion system, and one in Boston, Mass., using brine as a circulating medium. There are, however, a number of short pipe-line systems in connection with ice making and cold storage operating with more or less success.

The small machine has come to be a factor, and has reached a degree of perfection that makes it possible for any one who has daily use for from a half a ton refrigeration up, to secure an equipment that can be economically and successfully handled. Many thousands are now in use.

The domestic machine, as it is called in counter distinction to the small machine, has received a vast amount of attention, but the problems confronting the use of such a machine are enormous, and while many of them have been solved it cannot as yet be considered a commercial success for use in the average-sized family. There are, however, some half dozen machines designed for domestic use which have reached that degree of perfection where their further development will be watched with much interest.

Our engineers have taken up the question of ventilation and air purification in connection with refrigeration, and the improvements shown in cold storage and brewery installations testify to the value of the work they are doing. Air conditioning, as it is called, is more and more becoming a factor in human existence, as by means of refrigeration it is possible to obtain a humidity control which is so essential in certain lines of work and especially in buildings where people congregate. Refrigeration has been successfully used for the cooling of rooms in hospitals, banks, hotels, business houses, factories, theaters and churches.

Some of the best engineering talent has been engaged for a number of years in developing suitable insulation for cold storage houses so far as to minimize heat leakage into chilled spaces and to perfect material that would withstand moisture, lessen the fire hazard and not cost more than the limits of economic operation would warrant. Likewise, in order to avoid losses in the refrigerating apparatus, piping, etc., insulation of liquid and suction lines, of brine coolers and brine pipe, of freezing tank, water storage tanks, accumulators, receivers, etc., has been almost universally applied. Insulation as an economic necessity, wherever refrigeration is applied, has shown marked improvement both on the scientific and practical sides during the past twenty-five years.

The scope and usefulness of refrigeration, as applied to transportation of perishable fruits, have been greatly extended by the development of precooling. As pomologist for the United States Department of Agriculture, G. Harold Powell, about 1905, conducted experiments in California on precooling of fruits, which were so successful as to fully establish the value of this method. Since then large precooling stations have been erected at Roseville, San Bernardino and Colton, Cal., also in Texas and other Southern States. One of these California plants alone is able to precool at one time thirty-two cars of fruit in four hours.

Precooling, as is well known, consists in reducing the temperature of the fruit or product to be transported before shipment is begun. Two methods are in use—one cooling the goods in a warehouse before placing them in refrigerator cars, and the other cooling the goods after they have been placed in the car by circulating cold air. The fruit that has been precooled not only arrives at its destination in better condition, but it is not necessary to re-ice the refrigerator cars as often, thus saving time and expense in transit. Experiments with an aim to adapting precooling to a wider range of fruits, vegetables and other perishable foods are being conducted at the present time. Precooling has also made it possible to ship to much greater distances, and thus greatly extend the market for many small fruits, such as grapes, cherries, red raspberries and other highly perishable fruits.

SURVEY OF EXTENT OF REFRIGERATING INDUSTRY IN U. S.

A brief survey of the province of refrigeration in industry and the magnitude of the refrigerating industry may not be without interest. Mechanical refrigeration has invaded, or is being utilized in, almost every department of human industry. In the "Ice and Refrigeration Blue Book" are listed about 140 separate industries in which refrigerating machinery is employed. It need only be mentioned that it has entirely superseded the whole natural ice product in the ice supply in all Southern States, and is rapidly displacing it in Northern States. Thus at the present time fully sixty per cent of all the ice supply of Greater New York is manufactured ice, and in Boston, Maine and Canada manufactured ice is being introduced in increasing quantity.

Cold storage has revolutionized market conditions in all perishable foods, conserving the surplus of harvests, providing a remunerative market all the year around, and thus stimulating production, and giving to all consumers a full variety of foods at all times of the year regardless of seasons of production.

There are now approximately 1,000 general cold storage warehouses in the United States having cold storage space of about 250,000,000 cubic feet. If to this be added the cold storage rooms in packing houses and breweries, the figures will become, with 650 packing houses and 1,200 breweries, approximately 3,000 cold stores and artificially cooled space of at least 400,000,000 cubic feet. Then if to this be added the cold storage rooms of creameries, produce merchants and apple storers, at least 2,000 more must be added, and the number of cold stores will reach at least 5,000 and the cooled space approximately 500,000,000 cubic feet, and that still would not include the cold storage refrigerators in some 3,000 meat markets, 600 or 700 ice cream factories, 1,000 hotels, 400 restaurants, 200 club houses, 400 hospitals, 500 confectioners and bakers, 300 department and general stores, 300 groceries, 300 public buildings, asylums, prisons, etc., 200 fish-handling plants, not to mention artificially cooled departments in chemical laboratories, powder factories, government posts, etc. The value of the products kept from spoiling by refrigeration in these more than 12,000 refrigerated places has never been estimated, but it must be enormous.

The effect of utilization of refrigeration upon consumption, production and prices has been phenomenal and offers a striking example of the benefits derived from the introduction of mechanical refrigeration. Thus, for example, statistics presented at the U. S. Senate Committee Hearings in 1911 showed that in New York City the average yearly per capita consumption of eggs during the decade 1880-1890, with no cold storage, was 265. During 1900-1910, with ample cold storage, it was 318. According to data compiled by the U. S. Bureau of Statistics the per capita consumption of eggs in the seven leading cities of the United States was 396 in 1900, when cold storage first came into general use, while in 1913 it had already risen to 492 per capita. Still more significant is the fact that while increased consumption would naturally lead to advance in price, and while prices for other food products did advance, as everyone knows, the price of eggs has not advanced, the average wholesale price per dozen in New York City during the decade 1880-1890 (no cold storage) was 26¾ cents during November, December and January and 15¼ cents during April, May and June; for 1900-1910, when cold storage was general, the average price in April, May and June was 17½ cents, in November, December and January 21¾ cents for 8/10ths of the eggs and 29¼ cents for fresh gathered, which numbered not over 2/10ths of the total, so that the average wholesale price for the year was actually lower after the advent of cold storage than before. A similar showing is apparent in butter statistics. Although per capita receipts have decreased since the advent of cold storage, the average wholesale price for first quality butter was 8¼ cents less during the winter season in the decade 1900-1910 than in the decade 1880-1890, and during the summer only 1¾ cents higher, as shown in official statistics, pulished in "Ice and Refrigeration Blue Book." The latest statistics compiled indicate that at present approximately 30,000 refrigerating machines are in operation in the United States, representing an investment of approximately $350,000,000, and directly affecting products valued at over $3,000,000,000.

The economy gained through conservation by refrigeration of what was once wasted, and the added production made possible by cold storage of the seasonable surplus, have been of very much greater benefit than is ordinarily understood. When, for instance, it is considered how hundreds of millions of pounds of fresh meats are conserved by refrigeration so that fresh meat is obtainable at uniform prices and quality all the year around, and likewise many million pounds of fresh fish; that abundance of first class butter at all seasons is made possible only through refrigeration, and that before its advent rancid butter was the rule rather than the exception during eight months of the year; that but for refrigeration the 35,000 cars of citrus fruits from California and the 20,000 cars from Florida, which give to all parts of the United States an abundance of oranges and grape fruit at all times, would be impossible, and that the 5000 cars of cantaloupes shipped from the Imperial Valley of California alone this summer could not have been grown and marketed without refrigeration; that practically germ-free milk and certified pure milk for babies, which has had so great an effect upon the death rate and reduced sickness, would be impracticable without refrigeration and the refrigerating engineer; that the universal use of a variety of fresh foods all the year, made possible only by refrigeration, has added greatly to man's health, happiness and efficiency—when all these things are considered, when all the conservation by refrigeration is considered and all the added production because of the great widening of markets for perishable products is taken into account, it will be found that it is a conservative estimate to state that mechanical refrigeration serves to add at least one-half billion dollars to the wealth of the people of the United States each year, besides adding enormously to their comfort and convenience.

Many industries have appealed for help in obtaining "temperature below the normal for useful purposes," which is in part the definition of the field of the refrigerating engineer as enunciated by the first president of The American Society of Refrigerating Engineers—Mr. John E. Starr. And they have not appealed in vain, for the engineer has been ready to solve the problems offered, and in some instances to anticipate them and call the attention of the manufacturer to the good that they might get from the use of refrigeration.

Think of there being in the photographic art a refrigerating capacity in one factory alone of over 3,000 tons! And consider that refrigeration is now playing an important part in such diverse industries as silk manufacture, leather from hides to finished product, tobacco, dye stuffs, extracts, electrical apparatus, explosives, floriculture, horticulture, glues and gelatin, mercerizing, petroleum perfumes, rubber, shaft sinking, blast furnaces, automobile factories, laundries, paper manufacturing, the textile industry and many others.

The wonder is that in so short a time so much has been accomplished. Who could have twenty-five years ago in the wildest flight of imagination conceived such a development for refrigeration?

SOCIETY AFFAIRS

PROGRAM

Eleventh Annual Meeting

The American Society of Refrigerating Engineers

Monday, Tuesday and Wednesday, December 6, 7 and 8, 1915

UNITED ENGINEERING SOCIETY BUILDING
29 West 39th Street, New York, N. Y.

MONDAY, DECEMBER 6

Morning Session, 10 o'clock:

1. Roll call.
2. Reading of minutes of Tenth Annual Meeting and Fourth Western Meeting.
3. Report of The Council.
4. Report of Tellers of Election of Members.
5. Unfinished business.
6. New business.
7. Report of Tellers of Election of Officers.
8. Presidential Address—Louis K. Doelling, New York, N. Y.

Complimentary Luncheon, Engineers' Club, 12.30 *o'clock.*

Afternoon Session, 1.30 o'clock:

9. The Quincy Market Ammonia Relief Valve. F. L. Fairbanks, Boston, Mass.
10. The Flow of Superheated Ammonia Gas Through Orifices. Edward F. Miller, Boston, Mass.
11. A New Style of Steam Safety Valve. G. H. Clarke, Boston, Mass.

Evening Session, 8 o'clock:

12. The Testing of Thermal Insulators. H. C. Dickinson, Bureau of Standards, Washington, D. C.
13. Preliminary Measurements of the Specific Volume of Anhydrous Ammonia. D. R. Harper, Bureau of Standards, Washington, D. C.
14. Work of the Bureau of Standards on Constants of Refrigeration. H. C. Dickinson, Bureau of Standards, Washington, D. C.

TUESDAY, DECEMBER 7

Morning Session, 9.30 o'clock:

15. The Refrigerating Engineer. John E. Starr, New York, N. Y.
16. Present Status of Brewery Refrigeration. Peter Neff, Canton. Ohio.
17. Boiler Room Economies. F. E. Matthews, Leonia, N. J.

Subscription Luncheon, $1.50, *Engineers' Club,* 12.30 *o'clock.*

Afternoon Session, 1.30 o'clock:

18. Effect of Velocity and Humidity of Air on Heat Transmission Through Building Materials. J. A. Moyer, State College, Pa.
19. Equations for Ammonia Based on New Experimental Material, Frederick G. Keyes, Hoboken, N. J.
20. Steam Economy of an Absorption Refrigerating Machine. Clarence W. Vogt, Louisville, Ky.

WEDNESDAY, DECEMBER 8

Morning Session, 9.30 o'clock:

21. Temperature of Fruits and Vegetables in Transit in Refrigerator Cars. S. J. Dennis and W. E. Mosher, United States Department of Agriculture, Washington, D. C.
22. A Theory on Cooling Towers Compared with Results in Practice. B. H. Coffey and George A. Horne, New York, N. Y.
23. Recent Developments in the Study of Corrosion in Concrete Buildings and Pipe Lines. Morgan B. Smith, Detroit, Mich.

THE ANNUAL DINNER

A subscription dinner will be held at the Hotel Martinique, Broadway and Thirty-third Street, New York, N. Y., on Tuesday evening, December 7, 1915, at 7 o'clock. Subscription, $2.50 a cover.

After the dinner the following illustrated lectures will be presented:

Safety Pays. Dr. William H. Tolman, Director, The American Museum of Safety.

The Cork Industry. H. W. Prentis, Jr., Pittsburgh, Pa.

HOTEL HEADQUARTERS

Hotel headquarters will be at the Hotel Martinque, Broadway and Thirty-third Street, New York, N. Y. Engage accommodations direct, in advance of meeting.

PROCEEDINGS OF THE JOINT MEETING OF THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS AND THE AMERICAN ASSOCIATION OF REFRIGERATION, SAN FRANCISCO, CAL., SEPTEMBER 23 AND 24, 1915

The joint meeting of The American Society of Refrigerating Engineers and the American Association of Refrigeration was held in the Exposition Memorial Auditorium, San Francisco, Cal., on Thursday and Friday, September 23 and 24, 1915. This was the Fourth Western Meeting of The American Society of Refrigerating Engineers and a special meeting of the American Association of Refrigeration.

Two sessions were held, Thursday morning and Friday morning, both being presided over by President Louis K. Doelling, New York, N. Y. On Thursday afternoon the members and guests of both organizations attended the session of the section on refrigeration of the International Engineering Congress, which was presided over by Vice-President Theodore O. Vilter, Milwaukee, Wis.

Friday, September 24, was Refrigeration Day at the Panama-Pacific International Exposition, and the members and guests visited the Exposition during the afternoon, and in the evening enjoyed a splendid dinner at the Old Faithful Inn, within the Exposition grounds.

First Session, Joint Meeting, Thursday Morning, September 23

The first session of the meeting was called to order at about 10 o'clock A. M., and as more than a quorum for the transaction of business was present, President Doelling announced that the meeting was open for business.

A motion prevailed dispensing with the calling of the roll, as all present had registered.

The reading of the minutes of the Tenth Annual Meeting was also dispensed with, on motion made, seconded and carried, as the minutes had been published in the January number of the A. S. R. E. JOURNAL.

The report of the Tellers of Election of Members was next in order. It was as follows:

REPORT OF TELLERS OF ELECTION OF MEMBERS

NEW YORK, N. Y., September 11, 1915.

Mr. Louis K. Doelling, President,
The American Society of Refrigerating Engineers,
154 Nassau Street, New York, N. Y.

DEAR SIR:

Your Tellers of Election of Members met this morning and canvassed the ballots for new members closing September 10, 1915, and beg to report that the following applicants for membership have been elected to membership in the grades indicated:

MEMBERS

J. L. Baker, Omaha, Neb.
Charles W. Berry, Boston, Mass.
Ralph W. Bowers, Waynesboro, Pa.
Frederick T. Brandt, York, Pa.
Charles I. Day, Jacksonville, Fla.

MEMBERS—(*Continued*)

H. C. Dickinson, Washington, D. C.
William Earl Mosher, Washington, D. C.
Eric H. Peterson, Detroit, Mich.
George B. Stacy, San Francisco, Cal.
Edgar Steiner, St. Louis, Mo.
F. A. Wilcox, Dartford, Kent, England.

ASSOCIATE MEMBERS

C. Thomas Baker, Jacksonville, Fla.
E. F. Canfield, Winona, Minn.
Z. U. Dodge, New York, N. Y.
S. T. McAdam, Pensacola, Fla.
James Malcolm May, Wellington, N. Z.
George W. Mitchell, St. Louis, Mo.
Douglass F. O'Brien, Jr., Chicago, Ill.

JUNIOR MEMBERS

Harry C. Bruce, St. Louis, Mo.
Karl R. Dieterle, Los Angeles, Cal.
Frank B. Higgins, Canton, Ohio.
Stewart E. Lauer, York, Pa.
Stiles O. Lokey, Atlanta, Ga.
F. D. Petersen, Philadelphia, Pa.
Henry R. Rosebro, Jacksonville, Fla.
Guy V. Thompson, Portsmouth, Ohio.
Carl J. Weaver, Canton, Ohio.

Respectfully submitted,

(Signed) WILLIAM D. MONKS, *Chairman.*
PETER BINZEL, JR.
JEAN NEUBECKER.

President Doelling then declared the successful applicants for membership duly elected to their respective grades.

The next order of business was an address by Frank A. Horne, President of the American Association of Refrigeration, but as he was unable to be present the address was read by E. O. McCormick, San Francisco, Cal., a Vice-President of the American Association of Refrigeration. Before reading the paper Mr. McCormick welcomed the members and guests to San Francisco, and spoke of the many attractions in and around San Francisco and the State of California, also the Panama-Pacific International Exposition. Mr. Horne's paper was as follows:

ADDRESS OF FRANK A. HORNE, PRESIDENT OF AMERICAN ASSOCIATION OF REFRIGERATION

The American Association of Refrigeration greatly appreciates the opportunity of participation in a Joint Meeting with The American Society of Refrigeration Engineers, and in helping to emphasize the importance of refrigeration in its scientific and industrial aspects in connection with the Panama-Pacific International Exposition.

It is reasonable to expect as a result of this convention, the meeting of the International Engineering Congress, especially the session devoted to refrigeration, and the celebration of Refrigeration Day at the Exposition, that the interests we represent shall be decidedly profited and the general knowledge of the subject augmented.

It is fitting that the American Association of Refrigeration, including, as it does, the wider field in its work, should pay tribute to the strictly engineering and scientific side of refrigeration. Those who use refrigeration, and who have developed the industries which employ the process of artificial cooling, owe a great debt to the men who have worked out the basic principles and engineering details in the production of low temperatures upon which the superstructure of many enterprises and large industries rest.

On the other hand, the engineering profession requires the outlets which the demands of their customers call for in increasing volume. Those of you who attended the last meeting of the American Association of Refrigeration in New York, N. Y., last May, or have read the report of that convention, know somewhat of the things accomplished, the present activities and the future plans of the Association. The reports from the various committees and commissions indicated the progress of the organization, but also forecasted the future operations of our various departments of work. Some of the plans then proposed have been since put into effect, as, for example, the employment of a business manager. In addition to the investigations and work of the six regular commissions being similar to like commissions in other countries of the International Association, we have the following undertakings in prospect:

The Board of Engineers on Educational Work is now engaged in considering ways and means to carry out a project that is of the utmost importance to every one having investments in refrigeration and ice-making machinery; that is, the inauguration of complete courses in refrigeration in the many schools of technology not now giving them; to furnish capable men to address classes of such institutions on refrigerating subjects from time to time, and to address gatherings of operating engineers throughout the country.

The Committee on Papers and Lecures, in addition to suggesting topics for discussion or investigation and securing the preparation of papers to be read at the meetings of the Association, or to be used by those engaged in the proposed educational propaganda among universities, schools and operating engineers, have recommended the creating of a library in the office of the Secretary, to comprise all publications, articles and monographs that have been published in the English language, also all important foreign publications. This work has been commenced, and when a complete collection has been obtained, it is the intention, if sufficient funds are available, to have a complete bibliography made, published and distributed to all members.

The Commitee on State and National Investigations has contributed much valuable information on the effects of cold air on various foods. This field also appears to be a profitable one for development, and the Association will be kept advised of all such work done and will furnish information and data to members.

The Committee on Trade Extension has already obtained considerable data on exports of perishable foods, refrigerating machinery and supplies. Considerable interest has been aroused among many of our members, as the field appears to be a prolific one and susceptible of large development.

A feature that will undoubtedly appeal to every one is the American Association of Refrigeration Bulletin Service, designed to keep members informed of the work of the Federal Trade Commission in all matters pertinent to their interests in all branches of the ice and refrigerating industry.

A legislative bureau is also to be established in the Secretary's office by the time legislative bodies become active, and reports of all bills, national and State, inimical to the interests of the ice and refrigerating industry will be published from time to time in bulletin form.

It is the purpose of the Association to continue its relation with the United States Bureau of Standards through its Commission on Gases and Units, and from time to time publish and distribute the results of investigations under various commissions in bulletins such as have been already issued.

A vigorous campaign for new members has been begun for the purpose of increasing our members, so that our influence and resources shall be commensurate with the importance and size of the refrigerating industry.

It is a source of great regret that the writer cannot attend the joint meeting, and begs to convey to those present his best wishes for a successful convention and a pleasant and profitable visit to the Exposition.

Votes of thanks were extended to both Mr. Horne and Mr. McCormick for the very interesting address.

The next order of business was a paper by President Doelling on "The Oil Engine for Ice Plant Service."

The topic "Raw Water Ice Making Systems" was also discussed at this session.

The session then adjourned.

International Engineering Congress, Section on Refrigeration, Thursday Afternoon, September 23

The session of the section on refrigeration of the International Engineering Congress was called to order on Thursday afternoon, September 23, at 2.30 P. M., with Vice-President Theodore O. Vilter in the chair. The papers read before this section were as follows:

"Refrigeration in France." L. Marchis, Paris, France.

"Refrigeration in Sweden." Thor Andersson, Stockholm, Sweden.

"The Development of Refrigeration in the United States." J. F. Nickerson, Chicago, Ill.

The papers by Messrs. Marchis and Andersson were read by Joseph N. LeConte, Berkeley, Cal.

The session then adjourned.

Second Session, Joint Meeting, Friday Morning, September 24

The second and last session of the joint meeting was called to order at 10 o'clock on Friday morning, and the following two papers were presented:

"The San Bernardino Precooling Plant." C. M. Gay, Los Angeles, Cal.

"Rate of Heat Transfer in Double Pipe Brine Coolers." Van R. H. Greene and Fred Ophüls, New York, N. Y.

The topic "Power Required to Agitate Brine in Can Ice Tanks" was also discussed at this session.

A motion prevailed extending a vote of thanks to George H. Geisler, Oakland, Cal., Chairman of the Special Program Committee of the San Francisco Meeting, as well as the Committee, for the delightful entertainment provided.

President Doelling announced the following invitation from the special agent in charge of the exhibit of the Bureau of Standards at the Panama-Pacific International Exposition:

DEPARTMENT OF COMMERCE

BUREAU OF STANDARDS, WASHINGTON

The members of the AMERICAN SOCIETY OF REFRIGERATING ENGINEERS are cordially invited to visit the exhibit of engineering and physical work by the UNITED STATES BUREAU OF STANDARDS in the northwest section of the Palace of Liberal Arts, Panama-Pacific International Exposition. Engineers and physicists are requested to make themselves known to the attendants and learn how to avail themselves of the facilities of the Bureau.

ARTHUR W. GRAY, *Special Agent in Charge of Exhibit.*

Before closing the meeting President Doelling warmly thanked the members and guests for their attendance, and feelingly extended an invitation to those present to attend the Eleventh Annual Meeting of the Society in New York, N. Y., on December 6, 7 and 8.

The meeting then adjourned.

(Signed) W. H. Ross, *Secretary.*

AMONG THOSE ATTENDING FOURTH WESTERN MEETING WERE:

C. M. Gay, Los Angeles, Cal.
P. J. Coster, Lincoln, Neb.
Theodore Wetzel, San Francisco, Cal.
John S. Prell, San Francisco, Cal.
F. M. Bennett, Berkeley, Cal.
S. J. Scott, Visalia, Cal.
J. F. Nickerson, Chicago, Ill.
W. H. Ross, New York, N. Y.
H. C. Ruggles, Phœnix, Ariz.
Theodore O. Vilter, Milwaukee, Wis.
Louis Doelling, New York, N. Y.
George Geisler, Oakland, Cal.
W. P. Whittley, Seattle, Wash.
Henry Torrance, New York, N. Y.
E. O. McCormick, San Francisco, Cal.
J. Henry Beekman, Seattle, Wash.
Will P. Stevens, Los Angeles, Cal.
H. D. Bowles, San Diego, Cal.
W. H. Dohrmann, Brooklyn, N. Y.
J. V. Jamison, Jr., Hagerstown, Md.
Van R. H. Greene, New York, N. Y.
H. C. Wood, Cincinnati, Ohio.
F. F. Judd, Boston, Mass.
John N. Gruhl, Fresno, Cal.
W. S. Heger, Jr., San Francisco, Cal.
M. H. Robbins, San Francisco, Cal.
Norman Tyler, San Francisco, Cal.
Frank J. Toforo, San Francisco, Cal.
F. W. Langley, Venice, Cal.
Fred L. Williams, San Francisco, Cal.
Ramsdell S. Lasher, Evanston, Ill.
H. C. Parker, San Bernardino, Cal.
D. R. Merrill, Berkeley, Cal.
C. C. Handy, Berkeley, Cal.
W. F. Durand, Stanford University, California.
George B. Stacy, San Francisco, Cal.
J. B. Davidson, Davis, Cal.
A. H. Goldingham, New York, N. Y.
Joseph N. LeConte, Berkeley, Cal.

COUNCIL NOTES

The Council held a meeting in the office of the Society, 154 Nassau Street, New York, N. Y., on Wednesday afternoon, November 3, 1915.

The following officers and directors were in attendance, with President Doelling presiding:

Messrs. Doelling, Strickler, Matthews, Greene, Horne, Behn, Torrance and Ross.

The minutes of the meeting of the Council of August 12, 1915, were read and approved.

The Membership Committee presented applications for membership from the following, and they were approved in the grades indicated:

MEMBER

William I. Bodine, Springfield, Mo.

ASSOCIATE MEMBERS

Alan K. Gillespie, St. Louis, Mo.
Harry Hammond, Alexandria, Va.
Carl Nessler, Texas City, Tex.

JUNIOR MEMBER

Walter Jones, New Brunswick, N. J.

CHANGE OF GRADE OF MEMBERSHIP FROM JUNIOR MEMBER TO MEMBER

Carl E. Hoffman, Houston, Tex.

The expenses of W. H. Ross, Secretary, in attending the Fourth Western Meeting, San Francisco, Cal., amounting to $234.80, were approved and ordered paid.

The Nominating Committee, through its chairman, Henry Torrance, reported the following nominations for the offices falling vacant at the close of the Eleventh Annual Meeting:

President—Theodore O. Vilter, Milwaukee, Wis.
Vice-President (two years)—N. H. Hiller, Carbondale, Pa.
Treasurer—George A. Horne, New York, N. Y.

Directors (three years)—
G. F. Bein, Fort Worth, Tex.
H. N. Borgstedt, Yonkers, N. Y.
R. S. Broas, New York, N. Y.
H. S. Nulsen, New York, N. Y.
G. E. Wells, St. Louis, Mo.
Director (one year) to fill unexpired term of A. P. Criswell, deceased—
S. B. Carpenter, New Brunswick, N. J.

The resignation of Torato Takahashi, Tokyo, Japan, was accepted with regret.

The Program Committee reported progress in the preparation of program for Eleventh Annual Meeting. It also submitted list of papers to be presented. They are as follows:

Quincy Market Ammonia Relief Valve. F. L. Fairbanks, Boston, Mass.

The Flow of Superheated Ammonia Gas Through Orifices. Edward F. Miller, Boston, Mass.

A New Style of Steam Safety Valve. G. H. Clark, Boston, Mass.

The Testing of Thermal Insulators. H. C. Dickinson, Bureau of Standards, Washington, D. C.

Preliminary Measurements of the Specific Volume of Anhydrous Ammonia. D. R. Harper, Bureau of Standards, Washington, D. C.

Work of the Bureau of Standards on Constants of Refrigeration. H. C. Dickinson, Bureau of Standards, Washington, D. C.

The Refrigerating Engineer. John E. Starr, New York, N. Y.

Present Status of Brewery Refrigeration. Peter Neff, Canton, Ohio.

Boiler Room Economies. F. E. Matthews, Leonia, N. J.

A Theory on Cooling Towers Compared with Results in Practice. B. H. Coffey and George A. Horne, New York, N. Y.

Equations for Ammonia Based on New Experimental Material. Frederick G. Keyes, Hoboken, N. J.

Steam Economy of an Absorption Refrigerating Machine. Clarence W. Vogt, Louisville, Ky.

Temperatures of Fruits and Vegetables in Transit in Refrigerator Cars. S. J. Dennis and W. E. Mosher, United States Department of Agriculture, Washington, D. C.

Effect of Velocity and Humidity of Air on Heat Transmission Through Building Materials. J. A. Moyer, State College, Pennsylvania.

Recent Developments in the Study of Corrosion in Concrete Buildings and Pipe Lines. Morgan B. Smith, Detroit, Mich.

Safety Pays. Dr. William H. Tolman, Director, The American Museum of Safety.

The Cork Industry. H. W. Prentis, Jr., Pittsburgh, Pa.

The Secretary was authorized to extend to publishers, subscription agents and book dealers a discount of 10 per cent on subscriptions to the A. S. R. E. JOURNAL.

The meeting then adjourned.

W. H. ROSS, *Secretary.*

NOTICE

The Society subscribes for the Monthly Catalogue of United States Public Documents, which is the only publication giving a complete list of all the Government's publications as they appear. Since The Journal can find space to review only a few of those of the many valuable monographs and publications issued by the various departments that will be found of most general interest to its readers, it is to be hoped that members seeking information along certain lines will make use of this catalogue, files of which will be found in the Society's office. On request, the Secretary will undertake to advise a member what publications have appeared on a given subject during the current year, and at what price they can be obtained from the Superintendent of Documents, Washington, D. C.

WANTED

Anyone knowing the present address of any of the following members is requested to communicate with the Secretary:

E. L. FLORY
OSCAR DREYER
D. L. HOLDEN
S. R. PHELPS
CHARLES J. HACKETT

A STANDARD BASIS FOR FINDING THE FUEL ECONOMY OF STEAM-DRIVEN ICE MANUFACTURING PLANTS

The accompanying report of the Committee on Suggesting a Standard Basis for Finding the Fuel Economy of Steam-Driven Ice Manufacturing Plants was made at our Tenth Annual Meeting. The acceptance of this report was laid over until the Eleventh Annual Meeting this fall, in order that members and others may have an opportunity to offer suggestions as to enlargements or limitations. In the interim all members are requested to forward their suggestions to N. H. Hiller, Carbondale Machine Company, Carbondale, Pa., the chairman of this committee.

The report of this committee is of interest to practically every member of the Society, and it is hoped that they will avail themselves of the opportunity to cooperate with the committee in making these proposed rules as complete and workable as possible.

Report of Committee on Suggesting Standard Basis for Finding the Fuel Economy of Steam-Driven Ice Manufacturing Plants

Three members of this committee met, together with Peter Neff, who was President of the Society at that time, and after carefully considering the subject begs to submit the following report:

The phrase "fuel economy" is considered a misnomer, and the true efficiency of an ice-making plant should be gauged by the steam consumption of that plant, as evidenced by the weight of feed water used, and any guarantee made as to the efficiency of the plant should be based in terms of boiler evaporation. For example, if the boiler is evaporating 100 tons of steam per day, and the plant is making 90 tons of ice per day, then the efficiency of the plant would be expressed as "Ninety Per Cent."

It was then suggested that in order to determine the efficiency of a plant it was first necessary to determine its capacity, and, therefore, this committee has taken the liberty of formulating a set of rules for determining such capacity, these rules to be discussed and amended so as to form a standard basis for conducting tests of ice plants. The following are respectfully submitted:

1. *Preliminary Run*—There shall be a preliminary run of at least 10 days prior to the date on which the test is to be started, for the purpose of standardizing the work of the plant and getting it into satisfactory and normal condition before the starting of the test. During the last 72 hours of this preliminary run the ice shall be pulled regularly and at a rate not less than the capacity specified in contract.

2. *Duration of Test*—The test run shall follow directly after the preliminary run, and its duration shall be five days, or 120 consecutive hours.

3. *Condition of Plant*—At the time that the test run is started the conditions of the plant are to be carefully noted, especially the temperature of condensing water, temperature of brine and the condition of the ice in the tank. At the termination of the test, conditions should approach those at the

start as closely as possible, particularly in the temperature of the brine and the condition of the ice in the tank.

4. *Readings*—The readings are to be taken every 15 minutes, and are to consist of at least the following:

a. Temperature of atmosphere taken outside of the building in the shade.

b. Temperature of condensing water at condenser inlet and outlet and outlet from machine.

c. Meter feed water.

d. Temperature of brine in ice tank about midway between top and bottom.

e. Temperature of feed water entering boiler.

f. Boiler pressure.

g. High and low pressures of the refrigerating medium.

In addition to these readings any other temperatures, pressures and data that may be thought desirable by the parties at interest may be taken, but it is considered essential that the above readings be taken.

5. *Method of Pulling Ice.*—The ice is to be drawn at regular intervals throughout the 24 hours. For example, assuming that 100 blocks are to be the required quantity to be drawn in 24 hours, then this would necessitate pulling the ice at the rate of one block every 14.4 minutes, or two blocks every 28.8 minutes.

6. *Over Capacity*—This committee recommends that the contractor be allowed to overdraw capacity each day as much as he deems his plant capable of handling, providing that the temperature of the brine is not increased in the tank nor so long as the ice drawn is of full weight, and does not contain cups exceeding those described in Article 7. The determination of the capacity shall be based on the average of the ice pulled for the complete time of test.

7. *Weight of Blocks and Limit of Cups*—The can fillers to be adjusted so that the blocks pulled during the test shall not weigh less than 2 per cent under or more than 2 per cent over that specified in the contract as the standard weight per cake. It is suggested that the size of cups be limited to an opening that will hold not to exceed one quart of water. For the purpose of ascertaining the weight of the ice drawn, it is suggested that every tenth pull be weighed during the test period, and that an arithmetical average of same be taken as the average weight of the ice blocks throughout the test.

8. *Economy*—If it is desired to ascertain the economy of the plant, such a determination should be made during the period of the capacity test. This economy to be based on the pounds of ice produced per pound of dry steam delivered by the boiler, as evidenced by the feed water used.

9. *Standard Conditions for Economy*—This committee recommends that a boiler pressure of 100 pounds per square inch and a feed water temperature of 208 degrees F. be taken as a standard on which to base economy guaranty. The feed water used shall either be weighed or measured throughout the whole period of the test.

10. *Preparation for Test*—If the water temperature varies to any marked degree from that specified in contract, the amount of the allowance for such variation shall be determined, and the parties at interest shall agree upon such allowance prior to the starting of test.

It will be the duty of the purchaser to see that the apparatus is properly cleaned and made ready for test. All condensers, either steam or ammonia, and other cooling coils, shall be properly cleaned and scaled. The boiler is to be put in proper condition and thoroughly cleaned. Any insulation of steam or exhaust or other exposed pipes to be completed by the party furnishing the same under the contract. The purchaser will also furnish the scales or make such other arrangements as are necessary for weighing the boiler feed water and ice. All other parts necessary for making test, such as thermometers or pressure gauges not necessary for the ordinary operation of the plant, are to be loaned by the contractor and disconnected by him on completion of test. Thermometers and gauges used throughout this test to be calibrated before and at the completion of test.

11. *Leaks*—Any leaks in the steam, exhaust, blow-off or water lines that would have any effect upon the economy or capacity of the plant are to be properly repaired by the party furnishing the same before the test is started.

12. *Arbitration*—In case of controversy it is recommended that a committee of three arbitrators be formed. The contractor and purchaser to choose one arbitrator each, and these arbitrators to decide between them upon an umpire or third arbitrator, and any decision arrived at by this board of arbitrators shall be final. The expense of this board to be shared equally by the contractor and the purchaser, and all these arbitrators, in order to qualify, shall be members of this Society.

AMENDMENTS PROPOSED TO THE CONSTITUTION

At the last annual meeting the President, Henry Torrance, on recommendation of The Council, proposed the following amendments to paragraphs C 17 and C 20 of the Constitution, subject to discussion and such modification as may be accepted by the proposer:

(Words in italics indicate new matter; words in parentheses to be omitted.)

INITIATION FEES AND DUES

C 17. The initiation fee for membership shall be as follows:

For Members and Associates, $5.

For Juniors, $5. A Junior on promotion to any other grade of membership shall pay an additional fee of $5.

The annual dues for membership shall be as follows:

For Members and Associates *residing in the United States and Canada,* $15; *elsewhere, $10.*

For Juniors *residing in the United States and Canada,* $10; *elsewhere, $5* (for the first six years of their membership, and thereafter the same as for a Member or Associate).

THE COUNCIL

C 20. The affairs of the Society shall be managed by a Board of Directors chosen from among its Members and Associates, which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, the two Vice-Presidents, Treasurer, (and) fifteen Members and Associates, *and the last five living Ex-Presidents, ex-officio.* Seven members of The Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of The Council, but shall not have a vote therein.

These proposed amendments will come before the Eleventh Annual Meeting for final discussion and action.

A. S. R. E. Journal

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PUBLISHED BI-MONTHLY

BY

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

154 NASSAU STREET, NEW YORK, N. Y.

Edited by the Secretary, under the direction of the Publication Committee.

Reprints from this publication, which is copyrighted, may be made on condition that the full title of paper, name of author and date of presentation to the Society are given.

Subscription price, United States and dependencies, $3.00 a year; Foreign countries in Postal Union, $4.00 a year.

Entered as second-class matter at the Post Office at New York, N. Y., under the Act of August 24, 1912.

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OFFICERS AND THE COUNCIL OF THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

1916

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(Term expires at Annual Meeting of 1916)

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(Term expires at Annual Meeting of 1916)

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(Term expires at Annual Meeting of 1917)

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(Terms expire at Annual Meeting of 1917)

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(Terms expire at Annual Meeting of 1918)

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J. F. Nickerson, Chicago, Ill. . .
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Herbert S. Nulsen, New York, N. Y.

Committee on Standardization of Ammonia Fittings:

Thomas Shipley, *Chairman,* York, Pa.
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Ezra Frick, Waynesboro, Pa.
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A. S. R. E. Journal

VOL. 2 JANUARY, 1916 NO. 4

THE REFRIGERATING ENGINEER*

BY JOHN E. STARR, NEW YORK, N. Y.

I did not intend this to be a formal paper on the subject, but rather a friendly talk with you of the duties of the refrigerating engineer to himself, to his clients, or employers, to his contractors, if he is acting as a consulting engineer, and especially his attitude towards this Society. I am free to confess that, as to the last item, I have entirely rewritten my remarks.

As one of the grandpa's of this Society I was inclined to take a grandpa's prerogative and scold you about certain matters that I will allude to later on. But since I received the programme of the papers that have been and are to be presented to you at this meeting I have lost my grouch.

I intended to say that I feared this organization was drifting dangerously away from the ideals that were set up when it was organized, and that we were perhaps forgetting that our constitution states: "The object of this Society is to promote the arts and sciences connected with refrigerating engineering."

The reason I intended to say this is that for several meetings past there seemed to be more of a desire to exploit personal or special methods of doing things or, to be plain, to advertise, than to promote the arts and sciences connected with refrigerating engineering.

I have feared this tendency, for if it continued it would entirely lose for this Society its position as an authoritative body.

The discussion in our meetings of any subject from a standpoint based purely on the business interests of the parties taking part in the discussion, or the reading of papers designed to exploit a particular apparatus or material for business reasons only, belittles us as an authoritative body in the eyes of the public.

I intended to say that I had heard the criticism of outsiders that some of our meetings resembled more the meeting of an association of manufacturers than of engineers seeking the truth. I also intended to say that the loss of prestige as an authoritative body meant the loss of one of our main objects as an association, and hence a loss to every member of it.

As a former secretary of the American Society of Mechanical

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* Read at Eleventh Annual Meeting, New York, N. Y., December 6, 7 and 8, 1915.

Engineers remarked: "The Association has a weight in any utterance which is many times greater than the individual. The public recognizes that such utterance was made in an arena where, if it could have been intelligently questioned and differed from, it would have undergone that ordeal then and there." Had Mr. Hutton added the words "honestly and without prejudice," which, of course, he implied, the statement would have been stronger.

I had intended to quote, even at the risk of laying myself open to the charge of egotism, from the address that I had the honor to make as your first President in 1905. This quotation was as follows:

"In taking up our work, a certain sense of perspective is necessary. Most of us are business men, as well as engineers. We have our business interest and clashes of interests, our rivalries and jealousies, like other business men.

"In the consideration of principles, particular matters may be discussed by us that will touch one or the other closely. Our forum, however, must be a forum for all the membership and its discussions for the good of all. The truth is our aim, and the seeking of it our work. Within our doors a perpetual truce should prevail, and the sword and buckler of every-day strife is to be laid aside at our threshold."

This position I intended to reaffirm, and I intended to say some other things; in fact, I imagine I would have gotten myself disliked, but the Programme Committee for this meeting has arranged for a series of papers to be read here of a character worthy of our Society and of such importance to the "arts and sciences connected with refrigerating engineering" that I will gladly drop the first person singular, which has been so prominent in the preceding sentences, and was used only because it was intended to assume full personal responsibility.

It is hoped that nothing said will deter any member who really has some new and real contribution to the science of refrigeration from bringing it before the Society, even if it be a personal exploit.

It is contended, however, that anything brought before this meeting must be of a nature that will be of real benefit to the advancement of our objects, and that the time of our meetings should not be taken up with wrangling over the merits of rival methods of manufacture, nor, on the other hand, should the subject be treated in a manner so abstruse to the greater part of the membership as to not allow of discussion then and there.

A considerable part of the advantage of our meetings is in getting together and discussing particular points of individual interest outside of this room. In this room, however, it seems that our work should be solid and thorough on points of general interest to the arts and science of refrigeration, and our conclusions should be made to carry a weight backed by the Society as a whole.

Turning to the question of duties to your client or employer. The consulting engineer's duties are first and always to his client, but to be exercised in a spirit of fairness to the contractor.

If he is fit, as a consulting engineer, to perform his duties he should know how to draw his specifications and contracts in a way that will protect his client, and should not rely solely on the warranty of the contractor. Of what use is it to the client to draw up a set of specifications, detailing everything that the contractor shall do, down to the last valve or nipple, and then put in a rider binding the contractor to warrant the performance of the apparatus? He might just as well not have an engineer.

In this connection, I recommend a careful study of a valuable paper read by John C. Wait before this Society in 1906 and printed in Volume II of our Transactions, entitled "Contractors' Warranty versus Engineers' Specifications." The subject is treated in this paper from a legal and common-sense standpoint, and is well worthy of a careful review by most of you, whether engineers, owners or manufacturers.

Remember that the object of your client is to make money out of the plant you design for him, and that it is up to you to study broadly all of the elements that will contribute to this end. You will have to take into consideration, not alone the operation of his plant at peak load, but make the best possible estimate of his load-lines for the year.

Apparatus which would be absolutely the best to install in one case would be probably the worst in another case. Water and climatic and labor conditions, combined with load-line conditions, make it impossible for you to generalize. Each case must be taken by itself and must be considered as a business proposition in the light of your engineering experience.

Remember that while you have duties to your client you also have duties to your contractor. You will have to give the contractor a square deal. It oftentimes happens that you have a client who, either through ignorance or by reason of shady business methods, is inclined to take an unfair advantage over the contractor, and it is the duty of any self-respecting engineer to see that, as far as possible, justice is done to both sides.

It is also the duty of the consulting engineer to see that the contractor does not make warranties which the engineer knows he cannot carry out. There are irresponsible contractors who will promise to do most anything and agree to warrant it under seal. A contract to do the impossible, if the buyer knows it is impossible, is a void contract, and you are not benefiting your client, but, on the contrary, are injuring him if you permit any warranty in the contract which you know is absolutely impossible to be fulfilled.

Another thing to be remembered is that there is no engineer, no matter what may be his experience, who knows it all, and it is no disgrace to say that you do not know, when it is a fact that you do not know, and herein, sometimes, you are justified in asking a specific warranty from a contractor.

If a contractor should come to you and say he has an apparatus that will do a certain thing and will keep on doing it, and you do not know of your own knowledge or experience that he can do what he says, and you have not the time to ascertain, you are perfectly justified in asking him to warrant it, but remember always that this warranty must be under seal to make it binding.

You are, of course, justified at all times in demanding warranties as to workmanship and material, and there is, of course, no harm in putting into your specification and contract such warranties as the contractor is perfectly willing to give in any event which you are sure that can be performed.

As to the duties of a refrigerating engineer who is working for a manufacturer, whether of refrigerating machinery or carrying on refrigerating operations, there is, of course, but one rule, and that is: That as long as he is being paid by his employer, his sole aim should be to further in every honorable way the interests of his employer.

The next question, which often arises when an engineer changes his employment, is how much of the knowledge which he has gained in his previous employment he is permitted to divulge to his new employer. As a broad principle, it may be stated that legally, and perhaps morally, any information or experience which a man gains in his work belongs to him to do with as he pleases, but neither legally nor morally is he entitled to take with him the actual property of his former employer; that is, he is not entitled to have in his physical possession drawings, price lists or other papers which were made up during his previous employment and at the expense of his previous employer, but, it must be admitted as a practical and even as an ethical matter, that no employer can expect a departing employee to hold secret from anyone else any knowledge or experience such employee may have gained during the term of his employment.

This is the view of the law of the land, and the extent and nature of the detail and methods of the previous employer that may be divulged to a new employer lies largely in the conscience of the employee.

DISCUSSION

President Doelling—I believe you will all agree with me that we are very thankful to Mr. Starr for his paper. He has spoken in public what many of us have felt and spoken of privately. It is very gratifying to hear from Mr. Starr that he finds the conditions he mentioned in the first part of his paper have improved very materially. For this, I think, we have to thank our Programme Committee, which has worked very hard and very effectively to provide papers that are such as should be presented to an engineering society.

G. T. Voorhees—The sentiments Mr. Starr has brought forth, if they could be lived up to, and they might be, would be of the greatest value to all of us. I had the pleasure last night to hear Dr. Stratton tell how he secured so much good from the Bureau of Standards by insisting on the cooperation of his staff, and I took occasion afterwards to tell him how I wished that refrigerating engineers might find it possible to do likewise, through one means or another, with the idea of cooperating and working together for the same end. If the refrigerating engineers could cooperate they would find that, as they reviewed the experiences of a dozen different men, one man was eminently more qualified to handle a certain subject than a certain other man, and likewise with other subjects. As Mr. Starr correctly says, if petty jealousies could be overcome it would be of the greatest value to each other and the Society.

Peter Neff—I do not know whether all of the younger members of the Society have the early Transactions or not, but it will pay them to look up the first address delivered by Mr. Starr to this Society and read it through word for word. Mr. Starr has had the courage to quote from it, but these quotations by no means contain all the good information included in his first address.

EQUATIONS FOR AMMONIA BASED ON NEW EXPERIMENTAL MATERIAL*

By Frederick G. Keyes, Hoboken, N. J.

At the tenth annual meeting of The American Society of Refrigerating Engineers, a year ago, certain of the new experimental data obtained at the Research Laboratory of Physical Chemistry of the Massachusetts Institute of Technology by Robert B. Brownlee, Henry A. Babcock and the writer were presented. The investigation carried out consisted in determining with all the precision possible the vapor pressure of the liquid to the critical point, the specific volume of the liquid to pressures of 1,000 atmospheres and to 200° C., the specific volumes of the vapor in the superheated region and the specific heat capacity of the liquid. With the exception of the specific volumes of liquid ammonia under pressure, the data are all of interest to refrigerating engineers, since they serve as a basis for the computation of the various properties necessary in solving the problems which arise in engineering practice.

During the time which has elapsed since the former paper was presented all the protocols of the experiments have been re-examined. One readily perceives that in computing the results of experimental work many corrections must be applied before the final experimental values can be accepted. Reference in this connection may be made to the correction introduced to allow for the temperature expansion of the steel ammonia container and the mercury used to confine the ammonia, besides which the dilation of the experimental system itself, due to pressure, must be considered when reducing observations of the specific volumes of the ammonia at various temperatures and pressures. The various mercury and oil levels existing in the experimental system, of course, also enter into the reduction of each pressure observation. The calibration data of the thermometers, in itself a small investigation, are a factor requiring consideration before assigning a temperature value to a pressure or volume measurement.

This independent recomputation of the experimental values produced changes of only about one-tenth of one per cent in some of the pressure data. Part of the correction was due to the fact that the pitch of the thread used on the compressor screw had been changed previous to some of the final pressure measurements. The pitch of the old screw was assumed in computing one of the mercury levels within the apparatus during the later measurements, and this fact had in the first computations been overlooked. It was further found that in computing the volumes along one of the vapor isotherms at about 120° C. (248° F.) an incorrect value of the expansion due to the mercury had been employed. The rectification of this error affected the values of the first computation by about one-tenth of one per cent also. The assumption is not warranted that the data at best are better than within one-tenth of one per cent; if in spite of the pains taken they are indeed always as good as within one-tenth per cent of the true values. Experience teaches, however, that it is better to study and apply all corrections which in any way influence an experimental result.

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* Read at Eleventh Annual Meeting, New York, N. Y., December 6, 7 and 8, 1915.

In the case of the formula used to represent the specific heat of the liquid, a change of form was made which has been suggested from an examination of the variation of the specific heat capacity along the liquid saturation curve of other substances, for which data exist. The values of the specific heat capacities are, in consequence, a little different than given by the early formula at the higher temperatures above the range at which measurements were made.

The equation of state for substances in the vapor region has received an enormous amount of attention, and in computing the properties of various substances which are required by engineers a reliable equation of state is of first importance. In the paper appearing in The American Society of Refrigerating Engineers' Journal, November, 1915, computations were presented which compared experimental values with computated values of the pressure over a wide range of temperature for seven substances of diversified chemical constitution.

Steam, because of its importance in engineering work, has received perhaps more attention from experimenters and computers than any other substance. R. D. Mailey, at the Research Laboratory of Physical Chemistry of the Massachusetts Institute of Technology, devoted a number of years to the study of water in both the vapor and liquid phases, and for this substance the form of the equation of state has received a most complete verification. Ammonia is in its physical properties very similar to water. Thus water has a high dielectric constant, a high heat of evaporation, a specific heat capacity in the liquid phase roughly twice that of the vapor phase, while the coefficient $\left(\frac{\delta p}{\delta T}\right)_v$ in the liquid phase changes rapidly with the temperature. All these properties parallel the properties of ammonia, and on this account, as well as because of the success of the equation with the substances of more remote similarity, it was decided to make use of the equation for computing the properties of ammonia; in spite of the fact that the equations of state at present in use are of very different form.

Quite recently the ammonia tables computed by William E. Mosher on the basis of the old data have been published in Professor Goodenough's book.[1] It is expected that in the near future the ammonia tables based on the new experimental data will be issued and the purpose of the present communication is to present a discussion of the basis of these new tables, together with some comparisons of the new tables with the Mosher-Goodenough tables.

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[1] "Properties of Steam and Ammonia," John Wiley & Sons, 1915.

THE VAPOR PRESSURE OF LIQUID AMMONIA

Among the various physical properties of a substance, an accurate knowledge of the vapor pressure in relation to the temperature is very essential. While at first sight it would appear that accurate pressure measurements could be carried out with ease, it is found that when results reliable to one-tenth of one per cent are in question the experimental problem is far from simple. The establishment of a constant temperature bath, in which the temperature of the ammonia may be measured, the measurement of this temperature on some definite temperature scale, the purity of the ammonia sample and the attainment of the true equilibrium between the ammonia and its vapor are factors which must be under easy control before one can feel confidence in the reliability of the observed values. The establishment of the temperature scale is of great importance, and no doubt many discrepancies in vapor pressure data may be ascribed to a lack of adequate facilities for securing reliable thermometric readings. It is obviously logical to use the absolute scale, or what for practical purposes almost exactly approximates it; the constant volume hydrogen thermometer. Repeated and careful comparisons of the mercury thermometer used are necessary. This instrument at best is far inferior to a properly prepared pure platinum resistance thermometer, which may be calibrated under conditions which have been carefully investigated.

After passing through a two-years' experience at the Institute with mercury thermometers the vapor pressure temperature relations were redetermined using a sensitive platinum thermometer, the readings of which could be carefully related to the hydrogen thermometer scale.

The pressure measurements were carried out by weighing the force exerted against a steel piston floating in oil, which was in equilibrium with the pressure of the ammonia. This equilibration of the piston was given a prolonged study under various conditions and at various pressures. As a result of this work a new and sensitive method of determining the equilibrium position under a given pressure was developed, and by the use of this equilibrium indicating device the final calibration of the piston with a 40-foot column of mercury was carried out.

After more than two years' experience, during the course of which time a large number of measurements of the vapor pressure had been made under varying experimental conditions, a new apparatus was constructed and the work repeated. The final values derived from the final measurements were found to be in excellent agreement with the earlier.

The method of treating the vapor pressure data and calculating the constants of the equation which is to represent the vapor pressure is discussed in THE JOURNAL of this Society, already mentioned. The complete equation giving the pressure in atmospheres is as follows:

$$\log_{10} p = -\frac{1969.65}{T} + 13.31704 - 3.23858 \times 10^{-2}\, T + 5.4131 \times 10^{-5}\, T^2 - 3.2715 \times 10^{-8}\, T^3. \qquad (1)$$

Fig. 1 has been prepared by comparing the value of the Goodenough-Mosher tables with the new experimental values as calculated by the vapor pressure equation. It has been known for a long time that Regnault's high-pressure or closed monometer pressure readings were too low. Mosher's critical investigation resulted in increasing the pressure at higher temperatures. At temperatures below zero Regnault made use of a mercury thermometer, which seems to have caused an error in the temperatures observed. Table 1 gives the pressures in atmospheres assigned by Regnault and those resulting from the new work for each ten degrees centigrade from —30° to 100°.

TABLE I.—VAPOR PRESSURES OF LIQUID AMMONIA IN ATMOSPHERES

Temp. ° C.	Regnault.	New Values.
—30	1.140	1.177
—20	1.831	1.885
—10	2.821	2.892
0	4.189	4.278
10	6.018	6.127
20	8.402	8.535
30	11.45	11.60
40	15.26	15.44
50	19.95	20.16
60	25.62	25.90
70	32.47	32.78
80	40.59	40.95
90	50.16	50.56
100	61.36	61.83

THE SPECIFIC VOLUME OF SUPERHEATED AND SATURATED AMMONIA VAPOR

The saturated volumes of the vapor were all calculated by means of the equation of state for ammonia vapor. Ammonia vapor is peculiar in its property of absorbing very markedly on glass or other surfaces. It is this property which makes accurate measurements of the specific volumes very difficult. During the course of the Institute work it was found to be impossible to obtain consistent data for large volumes owing to this disturbing factor.

The error introduced by the absorption effect will depend somewhat upon the manner in which the measurements are carried out. For example, liquid ammonia may be condensed in a steel container at about —34° C. and the steel container sealed. Now at constant pressure and increasing temperature the volumes would approach

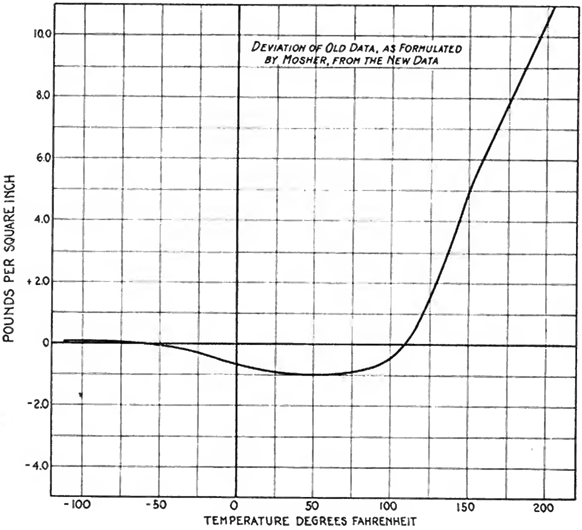

Fig. 1

more and more nearly the true values, while at constant temperature and increasing pressure the volumes would be increasingly too small. A further complication is introduced owing to the fact that the adjustment of the equilibrium of the absorbed ammonia takes time to establish itself. One observes that the state of the case just outlined holds only if the ammonia container has never been used to hold ammonia previously. If it has been used, it is practically impossible to rid the container of the absorbed ammonia except by heating in a vacuum to a high temperature. The consequence of this is that besides the definite quantity of ammonia introduced there may be in addition an unknown absorbed amount which later contributes to the volume of the absorbed ammonia even at comparatively high temperatures and low pressures.

Ninety-seven measurements were made in the course of the new work, and from a study of these data it was found that only at very large concentrations and high temperatures could reliable data be secured. The constants, accordingly, of the equation of state were determined from data in the region of high temperatures, and then by means of the equation comparisons were made by calculating the volumes of the measured pressures and temperatures for the data at large concentrations.

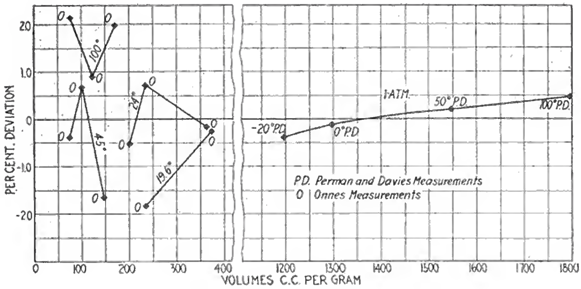

Fig. 2

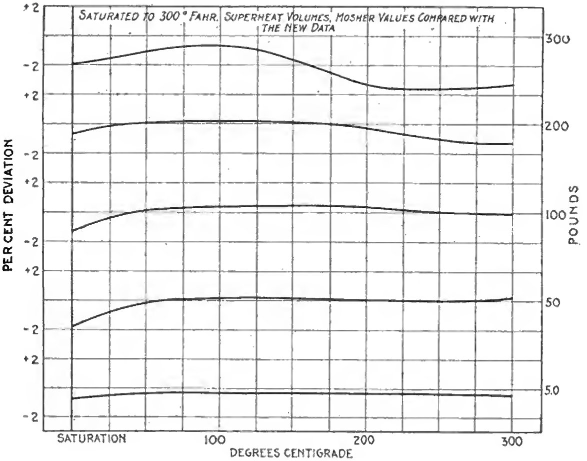

Fig. 3

A study of the deviations obtained from the comparisons confirms the conclusion that at least in the case of ammonia it is very difficult to avoid the absorptive effect. That the absorptive property of ammonia affected the results of other observers is scarcely to be doubted. Fig. 2 gives the per cent deviation of the specific volumes given by Onnes, and Perman and Davies from the calculations by means of the equation of state for the same temperatures and pressures given by these observers. The Onnes measurements are somewhat erratic in trend, but they indicate that at high temperatures the volumes tend to be too large, probably owing to the fact that the absorbed ammonia is given up when the experimental container is heated from room temperature to the high temperature of the measurement. It is evident that the zero line in any case is about as representative a line as could well be passed through the Onnes, and Perman and Davies data. Fig. 3 gives the differences between the values appearing in the Mosher tables at 5, 50, 100 and 200 pounds to 300° F. of superheat. The Mosher values are based primarily on the data of Perman and Davies.

The saturated values of the specific volume of ammonia vapor are also calculated from the equation of state, or, what amounts to the same thing, the volumes which result from the equation of state when the temperature, and the vapor pressure corresponding to this temperature, are substituted into the equation of state. Holst,[2] who, it appears, actually made the measurements reported by Onnes at the Third International Congress of Refrigeration in Chicago, 1913, used his data to evaluate the constants of an equation of state which then served for computing the saturation values of the specific volume of the vapor phase.

The Dieterici values and the Holst values have been inserted in Fig. 4, in which the heavy line represents the values calculated in the manner already described by the following equation:

$$p = \frac{R}{v - \delta} T - \frac{a}{(v - l)^2} \quad\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

R per gram per atmosphere $= 4.8177$.

$$\log \delta = 0.98130 - \frac{3.08}{v}.$$

$$a = 34610.$$
$$l = -1.173.$$

THE SATURATION SPECIFIC VOLUME OF LIQUID AMMONIA

The determination of the saturation specific volumes of the liquid do not require, for engineering purposes, to be established with the same precision as do the other properties of a substance. In the present instance the data due to Dieterici supply values which are quite accurate enough for all practical purposes. The Dieterici values needed, however, to be supplemented by measurements below zero and at temperatures approaching the critical temperature. New measurements were made at about —33°, 0°, and at intervals to 120° C. The former paper, already referred to, describes a method of treating liquid specific volume data in such a manner that, with comparatively few accurate data, it is possible to interpolate very accurately and thus calculate the specific volumes at any tempera-

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[2] Gilles Holst. "Les propriétés thermiques de l'ammoniaque et du chlorure de methyle." Leiden, 1914.

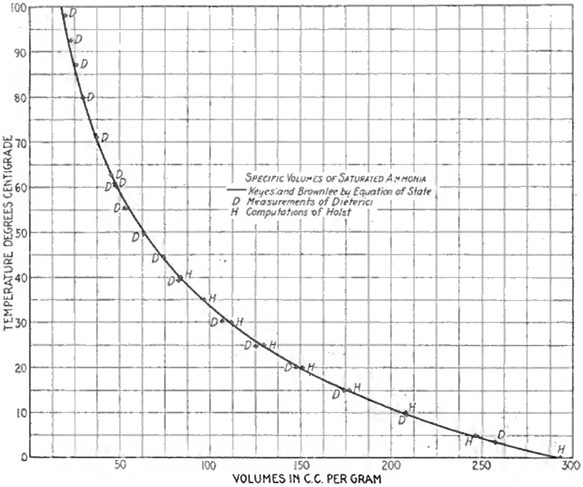

Fig. 4

ture within the range of the measurements. Table II gives the new values at each twenty °C. from —50 to 120, together with the values due to Dieterici, the older values due to Lange and those given by Mosher.

TABLE II.—SPECIFIC VOLUME OF LIQUID AMMONIA AT SATURATION POINT IN C. C. PER GRAM

Temperature °C.	Lange.	Dieterici.	W. E. Mosher.	Brownlee and Keyes.
—50	1.4375		1.4351	1.4222
—30	1.4895		1.4851	1.4742
—10	1.5480		1.5415	1.5330
0	1.5795	1.5656	1.5728	1.5656
20	1.6503	1.6342	1.6430	1.6387
40	1.7383	1.7227	1.7268	1.7255
60	1.8487	1.8250	1.8307	1.8328
80	1.9982	1.9595	1.9675	1.9757
100		2.1525	2.1684	2.1859
120			2.5538	2.5965

The critical constants are as follows:

$T_c = 132.9°$ C.

$p_c = 112.31$ atmospheres $\qquad \dfrac{R\,T_c}{p_c\,v_c} = 4.11.$

$v_c = 4.236.$

THE HEAT OF EVAPORATION OF LIQUID AMMONIA

Measurements involving the heat of evaporation of ammonia were made by Regnault, and of these measurements twelve[3] survived the Reign of the Commune and were later published. A careful consideration of Regnault's data involving the heat of evaporation of liquid ammonia has been given by Jacobus and Denton.[4] Franklin and Kraus[5] measured the quantity at the boiling point of liquid ammonia at atmospheric pressure (—33.2° C.), their value differing considerably from the value obtained by Estreicher and Schuerr.[6] The original communication containing the Estreicher and Schuerr measurements is not available, and hence a critical examination of the method used or a review of the data used in making necessary corrections is precluded. The method pursued by Franklin and Kraus consisted in vaporizing a definite volume of liquid ammonia at atmospheric pressure by supplying heat electrically. The calculation of the heat of evaporation requires the electrical energy, the density of liquid ammonia at—33° C. and the value of the calorimetric equivalent of the Joule at 15°. The value of the electrochemical equivalent of copper used by Franklin and Kraus was retained in the recalculation. The recomputed mean of the Franklin and Kraus measurements, using the latest density data and the 15° Cal. employed in the present tables, accordingly, is 336.58 Cal. at —33.2° C. A confirmation of the general correctness of this value may be obtained from the data concerning the elevation of the boiling point of liquid ammonia. The data in this connection is also due to Franklin and Kraus.[7] The mean value of *k* in Von't Hoff's formula,

$$\left(L = \frac{0.019885\, T^2}{k}\right)$$

taken from the elevation of the boiling point where water and alcohol were used as solvents, is 3.398—. The use of this equation leads to the value 336.8 Cal. as the heat of vaporization at —33.2° C. The value obtained by means of the Clapeyron-Clausius relation,

$$T \frac{dp}{dT} (v_1 - v_2) = L,$$

where the quantities on the left of the equation are obtained from the vapor pressure equation (1), the equation of state (2), and the equation of the liquid volume, lead to the value 336.5 at —33.2° C.

The equation relating the heat of evaporation of liquid ammonia to the temperature depends on the heats of evaporation calculated by means of the Clapeyron-Clausius equation from the data obtained at the Research Laboratory. The value of L was computed at 80°, 40°, 0° and at —70° C. The value of L at —70°, on account of the uncertainty of the vapor pressure equation at the lowest temperature, cannot be considered to possess the same relative accuracy as the values of the heat of evaporation computed for the higher temperatures. The values calculated at the temperatures mentioned were related to the temperature by means of a modified formula due to Thiesen. The Thiesen formula connecting the heat of evaporation with the temperature reads

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[3] Ann. de Chem. et de Physique (4) 24, p. 375, 1871.

[4] Jacobus, Trans. Am. Soc. Mech. Eng. 12, 307, 1891.

[5] Jour. Phys. Chem. 2, p. 555, 1907.

[6] Acad. Soc. Cracovie, Bull. 7A, p. 345, 1910.

[7] Am. Chem. Jour. 20, p. 811, 1898.

$$L = c\ C\ (T_c - T)^n$$

where c and n are constants and T_c the critical temperature. The equation satisfies the current ideas concerning the normal conditions of the curve (L, T)—namely, it yields a finite value of the heat of evaporation at the absolute zero and a zero value at the critical temperature. Taking logarithms of both sides of the equation, there results

$$\log L = \log c + n \log (T_c - T),$$

consequently the logarithms of L are a linear function of the logarithm of log $(T_c—\mathrm{T})$ and

$$\frac{d \log L}{d \log T} = n.$$

The equation was not found to hold strictly for liquid ammonia, although it does satisfy the values very nearly. To modify the equation it was assumed that the differential could be expressed as

$$\frac{d \log L}{d \log T} = a + b\ (T_c - T).$$

The resulting equation was then integrated, yielding the equation:

$$\log_{10} L = 1.56817 - 2.822 \times 10^{-5}\ (T_c - T) + 0.43387 \log_{10}\ (T_c - T) \ldots (3)$$

The values of L, calculated by the latter equation, together with the Regnault-Jacobus and Kraus values, are in Table III.

Gilles Holst,[8] in a recent publication concerning the properties of ammonia, computed the heats of evaporation of liquid ammonia from Regnault's data, using the more accurate Dieterici specific heat values now available in computing the corrections. The average temperature of Regnault's twelve measurements is 11.68° C., and the rate of change of the heat of evaporation with temperature may be taken from equation (3) to reduce each of Regnault's measurements to the average temperature. The result of this averaging is 295.7 Cal., whereas equation (3) leads to the value 294.3 Cal. at 11.68° C. The average Regnault value is, accordingly, 0.5 per cent higher than the value derived from the equation.

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[8] Gilles Holst. "Les propriétés thermiques de l'ammoniaque et du chlorure de methyl." Leiden, 1914.

TABLE III.—COMPARISON OF LATENT HEATS. Cal. per kg.

Temp. °C.	Calculated by Equation (3)	Observed.	Percentage Difference.
—33.4	336.7	321.3	—4.58 Estreicher & Schuerr
—33.2	336.5	336.6	+ .03 Franklin & Kraus measured.
—33.2	336.5	336.8	+ .08 Franklin & Kraus from ebullioscopic constant
—23.71	328.1	316.1	—3.6 Regnault-Jacobus-Denton.
—19.55	324.5	335.1	+3.2 Regnault-Jacobus-Denton.
— 9.72	316.0	317.0	+ .3 Regnault-Jacobus-Denton.
7.80	298.4	293.0	—1.8 Regnault-Jacobus.

Holst, in his summing up of the properties of ammonia, has computed the heats of evaporation by means of the Clapeyron Equation, using the saturated vapor specific volumes based on the Onnes "virial" equation of state and values of $\frac{dp}{dT}$ derived from a careful study of Regnault's vapor pressure measurements. Table IV contains the Holst values compared with the values computed by means of equation (3).

TABLE IV.—COMPARISON OF LATENT HEATS. Cal. per kg.

Temp. C.	Holtz.	Equation (3).	Percentage Difference'.
—40	328.5	341.98	—3.9
—30	322.5	333.62	—3.3
—20	316.0	324.86	—2.4
—10	309.0	315.74	—2.1
0	301.4	306.01	—1.5
+10	293.2	296.01	— .9
+20	284.24	285.49	— .4
+30	274.8	274.43	+ .14
+40	264.2	262.67	+ .67

Regnault's values of the vapor pressure were admitted to be low by Holst, and also the corrections applied by Regnault to his mercury thermometer below zero were somewhat in error. This circumstance would make uncertain values of $\frac{dp}{dT}$ resulting from the vapor pressure curve, and also the values of L calculated by means of the Clapeyron equation. A glance at the z diagram makes evident at once the erratic trend of the Regnault data.

Accurate experimental values of the heat of evaporation of liquid ammonia are necessary in the temperature interval between 0° and 20° C.; however, the Franklin-Kraus value at the boiling point and the mean Regnault value at 11.68 would indicate that equation (3) represents the heat of evaporation with substantial accuracy.

THE HEAT CAPACITY OF LIQUID AMMONIA

In carrying out accurate heat-capacity measurements by the method of mixtures it is necessary to have a large difference of temperature between the thermostat and the calorimeter, in order that the resulting temperature change may be sufficiently large to permit the necessary percentage accuracy. There is considerable difficulty involved in obtaining the precise value for the water equivalent of the calorimeter, and also in obtaining the value of the specific heat of the steel container or other receptacle containing the ammonia to be experimented on.

The following method of measuring heat-capacities was suggested by Dr. Charles A. Kraus, of the Research Laboratory: A steel bomb containing liquid ammonia under the pressure of its saturated vapor is brought to a constant temperature in a thermostat above the calorimeter; it is then dropped into a calorimeter containing a definite weight of water, and the temperature change of the calorimeter is observed. Another steel bomb identical with the first, but containing water under the pressure of its saturated vapor, is placed in the thermostat and dropped into the calorimeter. The weight of water in the water bomb is adjusted by repeated experiments until it gives practically the same temperature change as the ammonia bomb. This method does away with many of the objections to the method of mixtures. It is, of course, dependent for operaation on a large temperature-difference between the thermostat and the calorimeter, but the errors are the same, or nearly the same, for both the ammonia and the water experiments, and consequently compensated.

Mr. Babcock, after having completed the measurements of the specific heat capacity by the method already outlined, undertook the development of a method which would permit the measurement of the heat capacity over very small temperature intervals and at various temperatures approaching the critical temperature. The method consisted essentially in rotating a steel ammonia container suitably immersed in oil in a silvered Dewar tube. Measured amounts of electrical energy were introduced by means of a combined platinum resistance heater and thermometer. The amount of energy necessary to heat the oil and container was determined at a number of interval between 20° and 120°. The electrical energy necessary to raise the apparatus through one degree with the ammonia, after making the necessary corrections, is thus the difference between the two series of measurements at corresponding temperatures. These latter measurements have not been included in the equation used in the present tables, since the two more accurate values obtained by

Mr. Babcock cover the practical range of temperature required in engineering work.

The equation taken to represent the saturation liquid specific heat capacity is somewhat arbitrary. It seems probable that the heat capacity, because of the term $\frac{dv}{dT}$, becomes infinite at the critical temperature. For this reason the empirical equation chosen to represent the heat capacity was

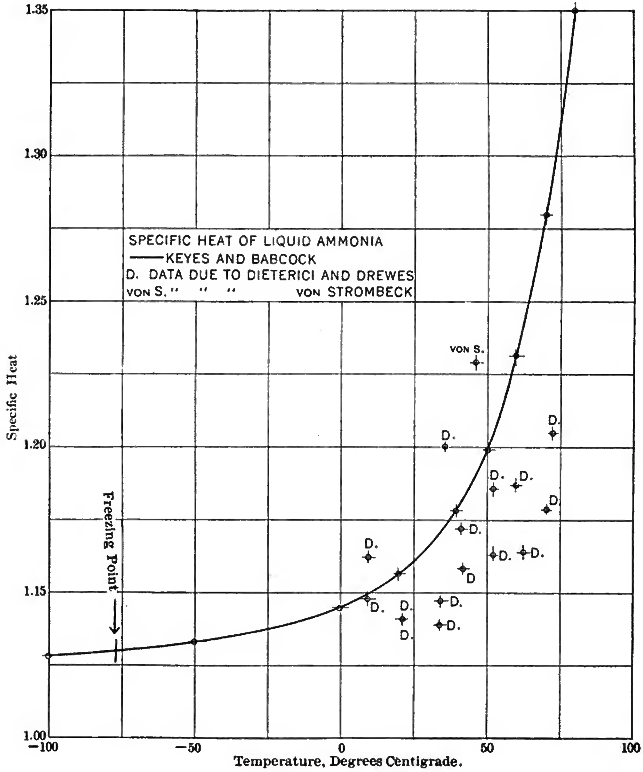

Fig. 5

$$C_{s_2} = 1.13747 - \frac{5.7575}{(T_c - T)} + \frac{898.53}{(T_c - T)^2}.$$

This equation passes through the two measurements made by Mr. Babcock. The course of the values is illustrated by Fig. 5, in which have been inserted the values reported by Dieterici. The equation for the integral heat referred to 0° C. is:

$$\int_{273.1}^{T} C_{s_2}\, dT = 1.13747\, T + 13.257 \log_{10} (T_c - T) + \frac{898.53}{T_c - T} - 345.556.$$

Table V gives a list of values as smoothed by Dieterici and reported in the Landolt and Bornsteine tables. The Dieterici values should be increased by about one per cent because of the calorie in which his results are expressed. The values given by other observers are also included for comparison.

TABLE V.—SPECIFIC HEAT OF LIQUID AMMONIA.

			Babcock
Drewes	0 C.°	.876	1.1450
Drewes	10	1.140	1.1501
Elleau & Ennis	10	1.02	1.1501
Drewes	20	1.19	1.1570
Ludeking & Starr	28	.886	1.1642
Drewes	30	1.218	1.1664
Drewes	40	1.231	1.1796
von Strombeck	45	1.229	1.1883
Drewes	50	1.239	1.1988
Drewes	60	1.240	1.2275

DISCUSSION

President Doelling—Mr. Mosher, you are very familiar with this kind of work; will you not give us your comment?

W. E. Mosher—I must express the same opinion about this work that I did about the Bureau of Standards' work; that is, I think both of these investigations are bringing us so much nearer the true values of the ammonia constants, more than anything available previously; it is a great step forward.

I would like to see Mr. Keyes' work published more in full, and would like to have access to the experimental data he has taken, and, in particular, a little more of the reasoning which has led him to the equation of state.

I think the Massachusetts Institute of Technology work and the Bureau of Standards' work will undoubtedly place us in position to use ammonia tables based on their work with great confidence. Moreover, the Society is certainly to be congratulated that it can have these original papers presented before it. It is only to be hoped that we can have them published more in full for the critical examination of the members.

H. Torrance—As to this question Mr. Keyes raises, about the kind of expansion that takes place when ammonia, liquid or otherwise, expands, it seems to me that the second equation,

$$\left(\frac{d\,T}{d\,p}\right)_Q = \frac{T\left(\frac{\delta v}{\delta T}\right)_p}{C_p}.$$

is the correct one. That is to say, in the actual case liquid is projected through the valve and thereupon expands.

G. T. Voorhees—I have this question rising in my mind often in connection with the new development of ammonia data: The engineer attempts to keep up to date with ammonia data and develops certain work along certain lines with the latest ammonia data, when suddenly newer data makes its appearance and the question rises in his mind as to whether he ought to retain the first data or use the new that comes along.

I am not sure but our position should be not to hope to work with extreme accuracy in this matter, that we should not, perhaps, change from our existing tables and methods until all these gentlemen have come to more or less agreement on what should be the basic values.

I have worked with the Mollier diagram, and I believe it is the most wonderful thing that has ever been gotten up to work with any refrigerant liquid, whether ammonia, carbon dioxide, or what not. I believe you can do more with it than any tables ever given to the world, and, having become somewhat familiar with the Mollier diagram of Mr. Mosher, I wish Mr. Keyes could tell us, in a few words, wherein his Mollier diagram may differ in any radical way from the Mollier diagram of Mr. Mosher, so that by such a short cut, and without spending too much time, whether there would be any material difference, whether it would be, for example, just a fraction of one per cent. If there is any material difference, and in spite of the fact that I have listened attentively, I do not think I have been able to gather just what that difference may be, and I wish we could be advised.

Mr. Keyes—Referring to Mr. Torrance's remark, I am by no means convinced that all the so-called throttling phenomena are in reality Joule-Thomson experiments. I was reading over last night the original account of Joule's experiments, and he made a large number of observations which imitate very closely the so-called throttling. Joule found that only under very particular conditions could he obtain numbers which coincided with the meaning expressed by the equation

$$\left(\frac{d\,T}{d\,p}\right) = \frac{T\left(\frac{\delta v}{\delta T}\right)_p - v}{C_p}.$$

The truth of the matter probably is that the reduction in the temperature produced by the expansion of a liquid through an orifice is something of a mixture, so to speak, of this latter equation and

$$\left(\frac{dT}{dp}\right) = \frac{T\left(\frac{\delta v}{\delta T}\right)_p}{C_p}.$$

At any rate, if one has a table prepared on the basis of either equation it is perfectly definite, and one may assume what one chooses with regard to the nature of the expansion process. I must confess that some further investigation is necessary to determine whether, in the so-called expansion valve, we really do have a Joule-Thomson phenomenon taking place.

The interesting thing to me about the new data of the properties of ammonia, when compared with Mr. Mosher's formulation of the old data, is the really comparatively small difference which exists. Differences to the amount in extreme cases of only a few per cent. This is a good sign. Engineers went through the same experience with the steam tables, as the early tables were considerably in error, while the successive tables which appeared came nearer and nearer together in values. The same thing is probably going to happen with the ammonia tables, and I feel that any new work will depart from the Institute values of ammonia but little.

Fig. 3 is intended as an example of what is meant. I have plotted per cent deviations of the new values from Mr. Mosher's values in the case of the superheated specific volumes, and you will notice that the per cent deviations are not wild. Two per cent is the amount of it in the worst case. The same is true of the vapor pressure curve, and I do not feel that any apprehension need be felt that any great change is going to be made in any of the properties of ammonia subsequent to this latter work.

W. E. Mosher—I would like to state that at the time Prof. Goodenough and myself prepared the tables there had been a great demand on the engineering experiment station for a set of workable tables for the use of engineers. A table was desired which gave increments of one pound and one degree, and it seemed logical at the time to utilize all experimental data available in the tables which were worked up. At about the same time this Society had a committee appointed to suggest a temporary standard table for use in the refrigerating industries of this country. Now, at the time we prepared the tables we knew that the investigation of the Bureau of Standards was in progress and also that the investigation at the Massachusetts Institute was in progress, and we were further informed that it would likely be five years before we would have available data which would serve as a basis for an authoritative set of tables. Therefore, in preparing the Illinois tables, the attempt was simply to make a summary after a critical study of the old data. I have pointed out, and Mr. Keyes has pointed out a number of times, that this was the entire object of those tables.

I have hoped that the tables might be adopted as a standard by the Society until such time as the Bureau of Standards can prepare standard tables for the United States. Undoubtedly the Bureau's tables will ultimately be the standard, and I believe the Goodenough-Mosher tables have become more or less standard, and thus they might simply fill in the interval until the Bureau of Standards' tables appear.

D. R. Harper—I have listened to this paper with a good deal of interest because it parallels so closely the work we are doing at the Bureau of Standards. I think the remarks made last night regarding the value of Mr. Keyes' experimental work need hardly be emphasized. We feel that much of his experimental work is of the very highest value and is going to add considerably to the certainty with which ammonia data is known. There are, however, a few remarks that I might make here, not by way of attempting to cast any reflection upon those values, but to point out to the members of the Society just the status of one or two of the features as they present themselves to me.

The interesting value referred to, which comes more fundamentally into the computations of the diagrams, is that of the heat of evaporation. The values are determined entirely by computation from equations based upon other data. If this data is sufficiently accurate the heat of evaporation derived therefrom will be sufficiently accurate for your purpose. Now, the specific densities in the saturated-vapor phase, which is one of the constants that come into this computation, is the constant the determination of which I dwelt upon last evening. Mr. Keyes made clear that, in so far as his computations are concerned, they are determined from measurements made in the temperature range above the ice point, and below zero he has taken the equation of state which he has been working with for all materials for so many years. In other words, the values are not taken for the lower range from direct observation, and in consequence this equation of state will have to be substantiated, I think. It seems to apply to ammonia very accurately throughtout the range where the measurements have been made. In fact, the observations seem to agree almost entirely with the equation of state; so I think you are entirely safe in taking the equation of state. It is, of course, an extrapellation, and a wide extrapellation; you are going down to very low temperatures, between zero and minus 50 degrees C., and that is a very dangerous thing to do. It should, therefore, be borne in mind that, while experimental data seems to substantiate the equation and to make that equation of state one to be generally accepted, we want to bear in mind what we are working with. Those are the thoughts that have been most prominently in my mind in going over this work on latent heat as computed from specific volumes which were not determined directly in the lower range of experimental data.

As regards the apparent agreement of some of the older measurements with the newer data, and as given in this paper, there are two or three questions I might ask Mr. Keyes. The agreement of the liquid heat of evaporation is determined from the rise in boiling point due to dissolved substances, and it occurs to me that this is rather doubtful as to meaning. What, for example, is the precision with which the data can be determined when the rise in boiling point is rather small. I will ask Mr. Keyes to state about what he thinks is the precision with which the rise in boiling point can be determined? If we did not obtain the agreement it need not worry us at all, as it is more or less accidental.

In the same way as regards the agreement of Dieterici's values of the specific volume of the vapor. Prof. Dieterici is a very careful worker, I think, but he was working under conditions difficult to control and with apparatus not well suited to his problem. I may be mistaken with reference to the accuracy with which he could determine specific volumes, but I think it is entirely accidental if he got less than 5 per cent error. The error might grow greater one time and smaller another, and by taking a large number of observations it may balance out to a slightly better average than 5 per cent, but so far as precision of his observations is concerned I doubt whether he could get an accuracy greater than 5 per cent. I was going to say 10 per cent, but perhaps that is too wide, so 5 per cent is better for me to give and, if Mr. Keyes will make a note, I would like to have him give his opinion as to whether he does not consider it little more than a happy accident that Dieterici's values happened to agree with those of the calculations given.

With respect to the agreement of latent heat as determined from specific volumes and vapor pressure change with temperature calculations, the values Mr. Keyes gives here involve also the question of the temperature scale. So far as you engineers are concerned, it makes no difference what temperature scale you use, because the difference between the hydrogen and platinum resistance scale is determined by various corrections, but when we come to using those differences for fundamental data, they may become significant; that is, if you are trying to determine the elevation of boiling point, the temperature scale becomes important. Mr. Keyes might indicate quite clearly what temperature scale he used. It will be of no particular interest to you for ordinary work, but it will be of interest to us, if he used a platinum resistance thermometer, what temperature scale he employed.

Those engaged in scientific work, we hope, will use the 20-degree calorie instead of 15-degree calorie. The Bureau of Standards has not taken any action toward officially recommending the 20-degree calorie, but the reasons for using it appear to us rather good. These reasons are set forth in one or two of our publications, and in so far as data is being expressed in units where there is a slight difference of practice throughout the world, it is very desirable to come to a standard basis and, after a careful review, we have accepted the 20-degree caloric instead of the 15. It is to be hoped there will some day be an international agreement on this matter.

Mr. Keyes—As to the temperature scale and platinum thermometer, I wanted to convey the idea that the constant-volume hydrogen thermometer is the one I referred my platinum thermometer readings to. This practically coincides with the absolute scale of temperature over the range in which measurements are carried out. Do you recollect, Dr. Harper, what the value of the 20-degree calorie is?

D. R. Harper—About one part in a thousand different from the 15.

Mr. Keyes—Dr. Harper's remarks about the heat of evaporation, as computed by the elevation of the boiling point, are quite apropos. We have a very small interval of rise in temperature to measure, and are perhaps measuring a few tenths of a degree. Now, it is obvious you cannot measure that with any great degree of accuracy. Dr. Kraus, however, is one of the most skillful of experimenters, and he determined the values of the elevation which I used in the computation Dr. Harper refers to. A good bit of his work is not published and is many years old now, but I should say that the value at 33.2 is perhaps 2 per cent or 3 per cent off, at the worst. By no means can we put as much confidence in it as we can in the data just referred to, but nevertheless it is of interest to compute the value in this manner. If it came out in the same order of magnitude, within 5 per cent, it would make one think that the more reliable values were somewhere near the truth.

The specific volumes, as given by Dieterici, cannot be subject to any high degree of accuracy. I think that Dr. Harper is very liberal in stating 5 per cent. I scarcely think they are as good as that at the lower temperature, and the agreement with the heavy line in Fig. 4 must be regarded as somewhat accidental. The final word on the specific volume of saturated vapor is, of course, given by the work of the Bureau of Standards. These values I have presented, be it emphasized again, are computed by an equation of state which I have good reason to think is a valid equation.

The equation of state rests on a rational basis, and I have talked about the equation to my friends for the last six years or so, but have not published anything in connection with it, except that which appeared in the A. S. R. E. JOURNAL of a year ago. I did at that time present some computations covering rather high pressures for a number of substances. If one examines those computations I think you will be struck with the fact that the equation does represent, with surprising accuracy, the p. v. T. relations of all the substances presented very well.

I have recently undertaken to review all the data in connection with the p. v. T. relations of air, and the equation reproduces the values to within about one-tenth per cent. In another investigation I have been taking up the equation of state for liquid ether, and the equation enables me to calculate the Bridgeman data to 12,000 atmospheres with an accuracy at that pressure of two or three per cent.

G. T. Voorhees.—I desire to ask a question of Mr. Keyes and Mr. Mosher, and will preface it by noting some work I did fifteen years ago, where I had to determine the properties of aqua ammonia ranging from pure anhydrous ammonia to pure water. In making these determinations, which, by the way, I suppose would be considered very crude compared with what the Bureau of Standards does, I found that it was absolutely impossible to work at the temperatures ordinarily given in using aqua ammonia, because the aqua ammonia would not exist at those temperatures, and for my own purposes I had to establish an arbitrary zero low enough so I could work with all the aqua ammonias. I worked around minus 30 degrees and referred all my work to that temperature.

In refrigerating work, I think there is a question of vital interest to this Society, and that is, why should we stick to the old arbitrary value of specific heat as being determined at zero C. or 32 F. in all our tables? It makes a rather inconvenient point to work from, in that we have to have plus and minus values of heat and liquids, etc. I am quite sure that, in any work we do, it would be more convenient if our starting point for heat of liquids started down where we are working from, at some temperature at least low enough that we are not going to get plus and minus values. I offer this as a suggestion to the committee that is working along these lines with the Bureau of Standards; that is, whether it is not advisable for this Society to suggest that the heat of the liquid be determined from some other temperature than 32 degrees, because it is not practical for our purposes. Minus 30 or minus 40 would serve better, or whatever the Society might set forth.

F. W. Frerichs—Why not start with absolute zero?

President Doelling—That would have to be taken up by the committee, I think; we cannot decide it here.

Mr. Keyes—The choice of 32 degrees F. as the reference point in tabulating the integral heat of the liquid is very proper when water is the substance in question. One notes that this is the freezing point of the substance, below which the liquid cannot exist. At any rate, cannot exist in equilibrium with the solid phase under the common vapor pressure. By analogy, the freezing point of ammonia should be chosen as the reference point for this substance. The freezing point is about — 77 degrees C.

A change in the units used could be well introduced along with reckoning the zero of integral liquid heat from the freezing point. English engineers persist in using the antiquated pounds, cubic-feet units when all the rest of the world have for many years turned to the use of a rational system of units.

SAN BERNARDINO PRECOOLING PLANT *

By C. M. Gay, Los Angeles, Cal.

ABSTRACT AND DISCUSSION

The term "precooling," properly applied, means the cooling of perishable products while loaded in the refrigerator car and prior to departure of train. It eliminates rehandling, and, since successful operation depends upon prompt cooling, it becomes a function of the transportation company, in the same manner that cold storage, before delivering to the railway company, is an unquestioned prerogative of the shipper. On the one hand, quick cooling is necessary to avert decay; on the other, the railroad company desires to accelerate the movement and availability of the cars. Once the advantages of precooling to the shipper, carrier and consumer were realized, the system was soon adopted on a large scale.

In 1907 the author was requested by the Atchison, Topeka & Santa Fé Railroad Company to study the possibility of cooling the contents of a train of refrigerator cars loaded with oranges, preferably in five hours or less, reducing the temperature of the fruit from the packing house temperature of 60 degrees to 90 degrees F. to the average refrigerator car temperature of 44 degrees to 48 degrees F. For the purpose of securing the necessary data, the company turned over its experimental refrigerating plant at San Bernardino, Cal. The information gained formed the basis upon which the present great Santa Fé ice making and precooling plant was built at San Bernardino in 1910.

Trains of cars from the surrounding citrus fruit district are taken to this precooling station as fast as loaded, and leave after a stop of only four hours, for their eastern destinations, with only occasional subsequent icing en route.

What makes precooling difficult is the fact that refrigerator cars, as built, are by no means air tight. Early experiments proved, that when a closed car with an inside air pressure corresponding to ½-inch water column was left standing, the loss of air from unseen openings amounted to 800 to 2,000 cubic feet per minute. The same leakage existed when an equivalent vacuum had been created within the car. Therefore, means had to be devised to equalize the pressure and volume of the refrigerated air entering and leaving the car, and to so balance the air circulating system between pressure and suction ducts that the point of equalization came as nearly as possible in the center of the car.

Plans are given, showing the size and arrangement of the buildings, insulation, air ducts, etc., built of reinforced concrete. The main building houses the power plant, ice tank room, day storage

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* Read at Fourth Western Meeting, San Francisco, Cal., September 23 and 24, 1915. Complete paper appeared in The Journal, September, 1915.

and winter storage. It is 511.5 x 132 feet. Connected to the winter storage are the precooler chamber and icing dock, extending for 737.5 feet. This gives a full length of 1,249 feet for the plant, or about ¼ mile. A concrete water reservoir, 80 feet square and 10 feet deep, is provided for the double pipe ammonia condensers. Water is obtained for the entire plant from an artesian well, supplying 500 gallons per minute. Another concrete reservoir, 20 x 30 x 10 feet, is used as a crude oil tank for the boiler plant.

Three 350 horsepower Sterling boilers are provided, also two 24 x 42-inch cross compound Corliss engines, each direct connected to one 300 ton, 17 x 34-inch duplex horizontal Vilter refrigerating machine. Adjacent to the engine room are filters and ammonia receivers, also a 36 x 24-foot water fore cooler tank.

The freezing tank room is 115.5 feet long, the full width of the building, and 21 feet high. There are three 75-ton freezing tanks, producing 300-pound blocks, three blocks being hoisted electrically at a time. One 300-ton brine tank is used for the precooling system and is equipped with Vilter flooded ammonia coils and an accumulator.

The capacity of the day ice storage is 900 tons, and of the winter storage, 7,000 tons for each of four sections. Below the floor of the day storage is a 49 x 33 x 11-foot brine chamber, containing a 200-ton double pipe brine cooler and three motor-driven brine circulating pumps.

The precooler chamber is 44.5 x 48 feet, insulated with 3-inch corkboard. It contains 2-inch galvanized brine coils. There are two sets of four double-inlet Sirocco blowers, each 10 feet in diameter, each set operated by one 85 horsepower motor. Either set can be used for blowing or exhausting.

From the precooler chamber lead two concrete tunnels or air ducts, one over the other, about 7 x 10 feet, the lower being the pressure duct, the upper the vacuum duct, both insulated. About 50,000 feet of ¼-inch galvanized brine pipe is arranged along the sides of these ducts. Over the ducts is the car icing platform. From the ducts lead 22-inch diameter galvanized steel spouts for connection to the refrigerator cars.

The plant has a maximum precooling capacity of 32 cars every four hours, or about 150 cars per 24 hours. The brine is circulated from the 300-ton brine tank through the refrigerating circuit and recooled in the brine tank. A large automatic equalizing valve connecting the pressure and vacuum ducts is placed near the terminals of the tunnels. An automatic intake valve is located at the outer end of the vacuum duct to compensate for air that might be lost when the warm air is blown from the cars.

The volume of air moved by the fans is about 260,000 cubic feet per minute. In the lower duct a pressure is held of ½ to ¾ inch water column above atmosphere, and in the upper duct a vacuum of ½ to ¾ inch of water. The air is delivered at 8 degrees to 10 degrees F. temperature and returns from the cars with 20 to 24 degrees F.

The flow of air through a car is at the rate of 6,000 to 8,000 cubic feet per minute.

Details are given showing the great facility with which flexible connection is made with the openings in the car roof.

The total amount of refrigeration consumed by a car of fruit varies from 1½ to 4 tons, average 3 tons, depending upon its initial temperature. About 1,500,000 cubic feet of cold air pass through a car, the average loss not exceeding five per cent. Cars are cooled to about 40 to 45 degrees F. After precooling the car is iced by means of the endless-chain conveyor.

The plant is fully equipped with facilities for measuring the quantity of air and brine in circulation. The original charge of ammonia was 13,000 pounds. The plant represents an investment of about $900,000.

Defrosting is done once a week by blowing warm outdoor air over the cooling coils, with the brine shut off. This takes 1 to 1½ hours. If desired, warm air can be blown through the cars to prevent frosting in severe cold weather.

At this plant a train of 30 cars can be connected to the cooling system by four operators within ten minutes. Since this plant has been established, over 28,000 cars have been precooled, and damage claims practically eliminated on shipments precooled by the Santa Fé Railroad.

DISCUSSION

C. M. Gay.—In 1907 the General Manager of the Santa Fé Refrigerator Dispatch came to me and opened up the question of precooling. He told me what I had not read and did not know of the experiments along those lines, and the wishes of the growers in the Southwest—I was then living in Los Angeles—and to a certain extent he outlined the necessity for precooling from a railroad viewpoint, as they then saw it. He was a man with a very full knowledge of transportation requirements, and I sat and talked with him for a long time trying to get his point of view. He said, "Is it going to be possible for us, Mr. Gay, to take a train of loaded fruit cars and cool them on the track within a period not exceeding four or five hours, ready for icing and shipping to the East?" I said, "I do not know. What experiments have you made?" He showed me their experimental plant. There was nothing very decisive about the plant, but it was well arranged for making experiments. After taking charge of the plant and making a few cursory experiments, I suggested one day that in order to precool a loaded car it was necessary to have a rack on the floor, so as to have a space under the fruit boxes for air circulation. He asked, "What will this cost per car?" I made a rapid calculation and said, "About $12 or $14 per car." He said, "Well, we will say $14, and for 14,000 cars it would make a total of $196,000. Well, you won't get it; that is out of the question."

Gentlemen, that was more education than I had ever got on that subject in one sentence. He followed that up immediately by saying, "I don't care what the station plants cost particularly, but don't you go to fooling with any of my cars, because every time you add a screw I have got to multiply it by 14,000." I don't think anybody had a better basis to work from than that. Accordingly, I dropped the question of making any changes in the construction of refrigerator cars, and this, probably, was an absolute necessity toward making fast progress.

In taking hold of this work I first undertook to determine the principle governing the circulation of air in refrigerator cars and found that every car used by the Transcontinental Refrigerator Lines was full of openings, unable to hold even a slight air pressure. As stated in the paper, this air leakage frequently ran as high as 2,000 cubic feet per minute under one-half inch water gauge pressure. With a vacuum on the car of one-half to three-quarters of an inch of water, I found there was a leakage from the outside to the inside of practically the same amount. I then came to the conclusion that a successful precooling plant must meet this condition of affairs, and that the work would have to be done with a minimum amount of deviation from atmospheric pressure in the car, and that a tremendous air circulation was necessary to accomplish the required refrigeration.

I investigated the specific heat of the fruit which was to be cooled, and developed the fact that the specific heat of oranges varies, according to the type and grade, thickness of skin, etc., from 0.73 to 0.82. I translated that into air units of heat, and found that I would have to have a very strong circulation of air in the car, and at very moderate pressures, to attain reasonable efficiency. With that as a basis, I proceeded to design the plant as described in the paper.

Henry Torrance.—Was the steam plant cheaper to operate than the electric? I see you have in use both steam engines and electric motors.

C. M. Gay.—The conditions that obtained at the time the plant was designed were peculiar. There was hardly anybody, except myself, who believed a precooling system, applied to a train of cars, was going to be a success, so that for auxiliaries it was much cheaper to install electric motors than it was to put in steam units for everything. Furthermore, if precooling should not prove to be a success, it would serve in the future as electrical reserve equipment. A very low rate for electric current was made at that time, so that the question of using steam for driving these auxiliaries was never really very seriously considered.

Besides, the main plant which is driven by steam, i. e., the ice making and ice storage plant is operated about 350 to 360 days per annum steadily, while the precooling plant, which is operated by electricity, is required for only six or seven months annually, with the possibility, *very insistently impressed on me at the time it was designed*, that it might never be operated at all. Consequently, I wanted to put as little expense as possible into that feature. That accounts for the operating of all the auxiliaries used in connection with the plant by electricity.

Van R. H. Greene.—What do you find the actual pressure drop through the car to be, between the inlet and outlet ducts?

C. M. Gay.—At the present time we maintain about two inches total differential, about one inch vacuum on the upper duct and an inch of pressure on the lower duct. That makes a two-inch difference to produce the circulation. We get an equalization of pressure at the car door. The length of all the laterals is the same no matter how twisted or turned, so that we have practically the same friction in both feed and return lines, the same control differential between feed and return, and we can absolutely maintain the point of equalization exactly at the door.

J. H. Beckman—In loading the cars, is it necessary to pile the boxes in any particular way?

C. M. Gay.—Yes, and in a way that had already been established by the necessity for standard refrigeration with ice by the various roads. They pile the boxes so many boxes wide, giving a certain amount of clearance between the rows of boxes, and they are also lathed across in tiers, which makes a space between the boxes both ways. That is necessary in order to cool off the car under standard refrigeration where the ice is placed in the bunkers, and it is admirably suited for the forced circulation used in precooling.

There is possibly one other point that I may give you which has developed during the past season. This is not from my own investigation, but that of the shippers. They took several refrigerator cars of the same series, so that a direct comparison of results could be made, loaded them with fruit in the citrus districts, placed ice in the bunkers, and shipped them, stopping at various stations between the regular icing stations, between San Bernardino and Kansas City. They also took a couple of cars and precooled them, stopping them at the same icing stations, and on arrival at Kansas City, made a note of the difference in amount of ice required on the cars that they precooled and those that were not precooled. They then took another batch of cars, precooled and iced them at San Bernardino and simply entered them at points where the other cars were iced, to find out at what point along the line, over 1,900 miles from San Bernardino to Kansas City, the temperature finally equalized between the car that was precooled and iced and given no further attention in the way of standard refrigeration, and the car not primarily precooled, which made the trip under standard refrigeration. The net result was that the car under the standard refrigeration traveled on an average of 1,400 miles before the temperature throughout the fruit reached the same temperature that the car precooled and iced without any further attention would reach. In other words, they had equalized at a point of 1,400 miles from loading station.

That means to the shipper and to the railroad company an extension of at least 1,400 miles of the safe shipping radius. After standard refrigeration was applied, there was, of course, little difference in temperature, even after it reached Chicago, at a distance of 2,100 miles, a difference of two or three degrees in favor of the precooled car, provided it had been given standard refrigeration reicing all the way through.

Henry Torrance.—That is 48 degrees?

C. M. Gay.—Yes, between 43 and 48 degrees, according to the season. My own observation and practice show that it is pretty nearly as easy to overcool fruit shipped under refrigeration as it is to undercool it. In other words, if you go very far below standard refrigeration temperature, the tendency is to precipitate moisture on the fruit, so that it is better to maintain the normal carrying temperature as nearly as possible, rather than try to reach extreme temperatures.

J. H. Beckman—What is the advantage of using the brine system, as compared to the direct expansion in precooling?

C. M. Gay—I am glad you have asked that question. The work of precooling is necessarily very irregular over any given period.

At the San Bernardino plant frequently thirty cars of fruit are thrown on the precooling circuit at once. If direct expansion is used, the maximum heat of these cars is felt at once throughout the precooling system, and back pressures go out of sight. During the first hour or more of precooling a complete setting of cars it is not unfrequent to take 10 degrees to 11 degrees of heat out of the fruit, while in the fourth hour of operation, upon the same line of cars, an average of three to four degrees' gain in temperature is all that can be accomplished.

With the brine system, the large volume of brine which is being circulated takes up the shock of the first hour or more of operation. When the operation of cooling one set of cars is finished, the machines are lowering the temperature of the brine and running along steadily until the next setting is completed. In this manner the total refrigerating capacity of the plant can be applied to precooling with the brine tank acting as an equalizer, storing up refrigeration between settings, and taking up the sharp heat of every fresh setting of cars without undue interference with the temperatures and conditions under which your refrigerating system is operating.

In my paper I have described the requirements and equipment of a plant designed for the precooling of oranges and fruits in loaded refrigerated cars, largely from a transportation standpoint, which without doubt would be the standpoint assumed by the average shipper situated within a reasonable radius of the railroad's precooling center. But there is another side to precooling requirements as viewed by the grower and shipper, or shipping organization, who have their own packing houses that handle several hundred cars of fruit each season.

In this case the shipper wishes to secure advantages extending beyond the preservation of fruit and longer radius of shipment, that is, the ability to hold a considerable quantity of fruit after precooling, to bridge over periods of overloaded market at the consumers' end and to facilitate shipments during rainy periods at the growers' end of the line, also to hold a sufficient stock on hand at all times to meet buyers' requirements in the various sizes and grades of packed fruit.

Several of the larger units of the shipping organization have installed their own plants, some of which are designed only for cooling and loading fruit, others are designed for cooling and loading the fruit, and at the same time manufacturing ice and furnishing icing facilities for the initial icing of the refrigerator cars, at the packing house when loaded.

A short description of the largest of these plants recently designed by and installed under the direction of the writer may be of interest.

On February 1, 1915, the precooling and ice making plant of the La Verne Orange Growers' Association, at Lordsburg, Cal., was put in operation. This plant has six combined precooling and storage rooms, with a guaranteed capacity of cooling 3,000 boxes of oranges per day to 36 degrees. The holding or storage capacity of the plant is 56 cars, or about six days run of maximum packing.

The arrangement of this house is such that the packed boxes are transferred to the precooling rooms by gravity conveyors, with no handling from the time packing and nailing of boxes is completed to the time they are in place in the precooler room. While being filled, and afterwards, each room is subjected to a cold air current controlled by graduated openings. The air current is usually kept at its maximum intensity until the fruit is cooled to 40 degrees F., and is then lessened until the regular storage temperature of 36 degrees F. is reached.

In loading out assorted cars, gravity conveyors are again employed to deliver the fruit to the power conveyor and elevator terminating at the car.

The cold air circulation is so arranged and controlled that there is no mingling of cold and warm air currents. Thus, the warm air strikes no cold surface to deposit moisture until it reaches the coil chamber, where the air is cooled and its moisture deposited on the coils.

At this plant a No. 8, double-inlet Sirocco fan, driven by 10-horsepower motor moves the required volume of air. The main air ducts are concealed beneath the floors and are well insulated with corkboard. The connection between the main air ducts and the openings of the rooms is made by utilizing a vertical space between the studding at the ends of the room, so that there are no air ducts in the way at any point, and the rooms are without inside projections to baffle the air currents.

At this plant the provisions for thawing off frost from the pipes are the same as those at the Santa Fé railroad precooler. Outside air is admitted through an opening in the fan suction duct and after it has passed over the frosted surfaces the air is discharged to the atmosphere. To confine the moisture the entire coil chamber is lined with galvanized iron on ceiling, walls and floor, with a drainage to the sump. From here the water may be diverted to the condenser supply or to the sewer.

A combined ice making and brine cooling tank is provided to supply brine for the cooling rooms and for ice production. A one-day storage room of 60 tons capacity, and a winter ice storage house of 2,000 tons capacity, the latter a first-class concrete structure with cork insulation, form prominent features of this plant.

The ice making tank has a capacity of 25 tons per day. The refrigeration for the whole plant is furnished by a 70-ton compressor driven by 100-horsepower, two-speed motor, operating the compressor at either 36 or 72 revolutions per minute, which feature adds to the convenience and economy, allowing the plant to be operated at half capacity in the early and late weeks of the season. A cooling tower for re-cooling the condenser water reduces the water expense to a minimum. One-fourth of the output of the freezing tank is clear, raw water—ice intended for the members of the Association and for local consumption.

The economical and profitable showing made by this plant during the season of 1915 prompted the growers of the adjoining district, known as the Indian Hills Association, to install a duplicate plant on their grounds, work on which is now progressing, and their plant will be in operation by January 15, 1916.

Both of the plants are driven by hydro-electric power obtained from public service corporations.

AMMONIA CONDENSERS

Reply by Fred Ophüls to a criticism of his paper on "Ammonia Condensers," read at the Tenth Annual Meeting, 1914, and appearing in the November, 1914, issue, Volume I, No. 1, of the A. S. R. E. JOURNAL.

The following is a translation of the review on the above paper, which appeared in the *Zeitschrift fur die gesamte Kaelte-Industrie*, July, 1915, No. 7, page 76:

> "The author has conducted a great number of tests on eight condensers of various types and sizes for the purpose of securing data to find the law which governs the rate of heat transfer in this apparatus. The article is very well written, and one must agree with the statements made regarding the conditions obtaining in the various phases of a condenser. Unfortunately, an error has crept in at the start of the mathematical analysis which renders the results useless. Using the author's notation, the following equation applies to the heat transfer of a condenser:
>
> $S \times k_m \times t_m = s_s\, k_{ms}\, t_{ms} + s_l\, k_{ml}\, t_{ml} + s_f\, k_{mf}\, t_{mf}$
>
> If, according to Ophüls,
>
> $$t_m = \frac{s_s\, t_{ms} + s_l\, t_{ml} + s_f\, t_{mf}}{S},$$
>
> it does not follow that at the same time
>
> $$k_m = \frac{s_s\, k_{ms} + s_l\, k_{ml} + s_f\, k_{mf}}{S},$$
>
> on the contrary, k_m is to be derived from the first two equations.
>
> "We regret that the data submitted are not complete enough to permit correcting the results of the investigation.—M. KRAUSE."

The impression received from the above review is that the results obtained from the test figures are incorrect and need recalculation. However, this is not the case, for either one of the last two equations can be used in the analysis in connection with the basic equation.

On checking up the calculations for the values of k_m, which are given for some of the tests in Tables 5 and 6, it will be found that only the equation giving the value of k_m was used. Equation 7 shows how the value of k_m can be found from the test results without finding any value for t_m. It will also be noted that in no part of the paper the combined mean temperature difference t_m was used. For this reason it would be better to eliminate equation 5 from the paper, or else add this explanation to make clearer the method of analysis used.

I am greatly obliged to Mr. Krause for having called attention to this discrepancy, but his conclusion that the calculations are incorrect on this account is wrong for the reasons above stated.—FRED OPHÜLS.

SOCIETY AFFAIRS

PROCEEDINGS

Eleventh Annual Meeting

The American Society of Refrigerating Engineers

NEW YORK, N. Y., DECEMBER 6, 7 AND 8, 1915

The American Society of Refrigerating Engineers held its Eleventh Annual Meeting in the United Engineering Society Building, 29 West 39th street, New York, N. Y., on Monday, Tuesday and Wednesday, December 6, 7 and 8, 1915. Six sessions were held, Monday morning, afternoon and evening, Tuesday morning and afternoon and Wednesday morning, all being called to order and presided over by President Louis K. Doelling, New York, N. Y.

This was the most successful meeting in the history of the Society. The attendance was 188 members and guests. Nineteen papers of unusual interest and value were presented. The large number of papers and important matters that came before this meeting of the Society necessitated continuing the meeting until late in the afternoon of the third day.

First Session, Monday Morning, December 6

The first session of the meeting was called to order at about 10 o'clock on Monday morning, December 6, and, as more than a quorum for the transaction of business was present, President Doelling announced that the meeting was open for the business before it.

As all members and guests had registered, the calling of the roll was dispensed with, as was also the reading of the minutes of the Tenth Annual Meeting and the Fourth Western Meeting, as they had been printed in the A. S. R. E. JOURNAL.

The report of The Council was the next order of business. It is as follows:

REPORT OF THE COUNCIL

The Finance Committee report shows all of the receipts and disbursements of the Society during the year. It is as follows:

REPORT OF FINANCE COMMITTEE

NEW YORK, N. Y., December 4, 1915.

Mr. Louis K. Doelling, President,
The American Society of Refrigerating Engineers,
154 Nassau Street, New York, N. Y.

DEAR SIR:

Your Finance Committee begs to report the following receipts and disbursements from November 28, 1914, to December 4, 1915:

Balance on hand November 28, 1914			$201.89
RECEIPTS			
Dues		$4,370.12	
Initiation fees		185.00	
Emblems		36.00	
Transactions, reprints, etc.		223.18	
Dinners		317.50	
Telephone		29.53	
The Journal—			
Advertising	$2,114.61		
Subscriptions, etc.	212.01	2,326.62	7,487.95
			$7,689.84
DISBURSEMENTS			
General printing		$554.25	
Emblems		29.25	
Dinner and luncheons		592.70	
Rents		637.58	
Salaries		2,272.51	
Library, office equipment, etc.		57.90	
Telephone		66.99	
Petty cash		512.40	
American Association of Refrigeration		10.00	
International Engineering Congress		15.00	
Premium on Treasurer's Bond		2.50	
Stenographic report, Fourth Western Meeting		60.00	
Stenographic report, Tenth Annual Meeting		115.00	
W. H. Ross, expenses, Fourth Western Meeting		234.80	
Miscellaneous		10.00	
The Journal—			
Composition, press work, binding, wrapping, paper	$1,590.50		
Illustrations	582.79		
Binders	297.01		
Mailing boxes for binders	10.54		
Wrappers for The Journal	18.94	2,499.78	7,670.66
Balance on hand			$19.18

Respectfully submitted,

(Signed) Theo. O. Vilter, *Chairman.*

(Signed) Geo. A. Horne, *Treasurer.*

The Council held three executive sessions during the past year, with your President presiding at each meeting. The dates on which and places where these meetings were held are as follows:

January 29, 1915, at New York, N. Y.
August 12, 1915, at New York, N. Y.
November 3, 1915, at New York, N. Y.

The Council respectfully presents herewith a report of the business transacted at its meetings, along with other matters that should be of interest to the membership.

At the January meeting, W. H. Ross was reappointed Secretary of the Society for another year, subject to the wishes of The Council, and on the conditions of contract of employment dated November 2, 1914.

At the January meeting an appropriation of $550 was made to cover the rent of the Society's office for the year 1915, as well as an appropriation of $390 for part services of an office stenographer.

The Year Book for 1915, of which 500 copies were printed, contained two lists of the members, one arranged alphabetically and the other geographically, in addition to other matter that has not appeared before in the Year Book. One copy was mailed to each member and the remaining copies were used to promote the membership of the Society.

At the January meeting, Ex-President Torrance was appointed a Director to fill the unexpired term of Ezra Frick, who resigned to accept the Vice-Presidency of the Society, and at the August meeting George H. Geisler, Oakland, Cal., was appointed a Director until the annual meeting of 1915, vice A. P. Criswell, deceased.

At the January meeting the prices per year for a subscription to the A. S. R. E. JOURNAL was set as follows:

Members of the Society in all countries, $3.00 per year.

Non-members in the United States and dependencies, $3.00 per year; foreign countries in the postal union, $4.00 per year.

Arrangements have been made with the postal authorities for the admission of the A. S. R. E. JOURNAL as second-class matter, providing the following words are added to Section C 17 of our Constitution, immediately following the last word of paragraph 4 of this section. Paragraph 4 now reads:

"The annual dues for membership shall be as follows":

The postal authorities suggest that it read:

"The annual dues for membership shall be as follows, including $3.00 for a subscription to the A. S. R. E. JOURNAL for one year":

To overcome the charge by the postal authorities for handling the A. S. R. E. JOURNAL at the third-class and parcels post rates, and that we may secure the cent a pound rate charged for second-class matter, it will be necessary, if these rates are to be secured now, as well as on the last five editions of The Journal, to suspend by motion the Constitution and amend this section, instead of proceeding in the usual manner. We trust such action can be taken at this meeting, as it will save the Society about $100 a year in postage, in addition to about $50 now on deposit.

It is apparent that The Journal, with binder, is a great improvement over our Transactions. The Society and membership are better able to keep in touch with each other through The Journal than was possible through the Transactions. So far not a single member has indicated a preference for the Transactions.

It is pleasing to note from the report of the Finance Committee that The Journal is already self-supporting, and in a year or two should bring an additional revenue to the Society, although it is not the purpose of The Council to compete in any way for advertising patronage to the exclusion of our trade publications, which have so generously cooperated with us in the development of the Society.

In addition to the standing and regular committees your President has made the following additional appointments during the year:

N. H. Hiller, Carbondale, Pa., was appointed to represent the Society at the Second Pan-American Scientific Congress to be held in Washington, D. C., on December 27, 1915, to January 8, 1916.

Tellers of Election of Officers:

John R. Livezey, *Chairman*, Philadelphia, Pa.
Donald Cole, Detroit, Mich.
Nelson J. Waite, Cleveland, Ohio.

Frederick L. Pryor, Hoboken, N. J., was appointed to represent the Society at a meeting of the Joint Committee on Classification of Technical Literature.

The Membership Committee presented applications for membership from the following, and they were approved in the grades indicated:

MEMBERS

J. L. Baker, Omaha, Neb.
Charles W. Berry, Boston, Mass.
Ralph W. Bowers, Waynesboro, Pa.
Frederick T. Brandt, York, Pa.
Charles I. Day, Jacksonville, Fla.
H. C. Dickinson, Washington, D. C.
William E. Mosher, Washington, D. C.
Eric H. Peterson, Detroit, Mich.
George B. Stacy, San Francisco, Cal.
Edgar Steiner, St. Louis, Mo.
F. A. Wilcox, Dartford, Kent, England.
William J. Bodine, Springfield, Mo.

ASSOCIATE MEMBERS

C. Thomas Baker, Jacksonville, Fla.
E. F. Canfield, Winona, Minn.
Z. U. Dodge, New York, N. Y.
S. T. McAdam, Pensacola, Fla.
James Malcolm May, Wellington, N. Z.
George W. Mitchell, St. Louis, Mo.
Douglass F. O'Brien, Jr., Chicago, Ill.
Alan K. Gillespie, St. Louis, Mo.
Harry Hammond, Alexandria, Va.
Carl Nessler, Texas City, Tex.

JUNIOR MEMBERS

Walter Jones, New Brunswick, N. J.
Harry C. Bruce, St. Louis, Mo.
Karl C. Dieterle, Los Angeles, Cal.
Frank B. Higgins, Canton, Ohio.
Stewart E. Lauer, York, Pa.
Stiles O. Lokey, Atlanta, Ga.
F. D. Petersen, Philadelphia, Pa.
Henry R. Rosebro, Jacksonville, Fla.
Guy V. Thompson, Portsmouth, Ohio.
Carl J. Weaver, Canton, Ohio.

CHANGE OF GRADE OF MEMBERSHIP FROM JUNIOR MEMBER TO MEMBER

Carl E. Hoffman, Houston, Tex.

The resignations of the following members have been accepted with regret:

J. Lathrop Gray, St. Louis, Mo.
D. L. Fagnan, New York, N. Y.
George W. Wright, Baltimore, Md.
George Berna, Ithaca, N. Y.
H. M. Moore, Des Moines, Ia.
Samuel R. Frantz, Waynesboro, Pa.
J. Spinka, St. Louis, Mo.
J. C. Bertsch, New York, N. Y.
T. Takahaski, Tokio, Japan.

The Council regrets to report the death of the following members during the past year, and it hopes suitable resolutions will be offered on their life work during this meeting:

A. P. Criswell, Chicago, Ill.
Benjamin F. Daly, Rockaway Beach, N. Y.
James Edwin Quigley, Pittsburgh, Pa.
William J. Webber, Boston, Mass.

Mr. Criswell was a Director of the Society at the time of his death, and your President fitly remembered his services to the Society by sending a floral tribute as an expression of the esteem in which he was held by the Society.

Another pioneer in the refrigerating field, Thomas L. Rankin, passed into the Great Beyond during the past year. Mr. Rankin was born in Ohio on June 16, 1839, and died at his home at Sacket Harbor, N. Y., on November 12, 1915. With the early history of refrigeration in the United States the name of Thomas L. Rankin was closely associated, and the effect of his natural trait of investigation and invention is still felt in the industry to which he gave so many years of his life.

A suggestion has been made to hold our annual meetings in some hotel and save the expense of meeting rooms in the United Engineering Society Building. This matter has received the attention of The Council, and it feels it is best to hold the meetings in the United Engineering Society Building. The charge for meeting rooms in this building costs us from $50 to $75, and this year it will be a little more. However, an expression from the membership on this question at this meeting would be welcomed by The Council.

Other suggestions have been the placing of a time limit on presentation of papers, as well as discussions, and that more time be allowed for discussion of papers and topics by either reducing the number of papers presented or that more sessions be held. These suggestions have been referred to the Program Committee with power to act, and the Program Committee reports that these suggestions were considered in making up the program for this meeting.

The Council is always pleased to receive suggestions from the membership, and the more the better, as this indicates an increased interest by the membership in the Society and its management. Every suggestion receives careful and courteous consideration.

The Council has endorsed the movement launched by the American Association for the Adoption of the Metric System, to educate the American public in the advantages of the metric system of weights and measures and to extend its uses in the United States. The Council feels that the metric system, by universal adoption, will afford the peoples of all nations a better means of communication, particularly of engineering data, than that of any other system proposed.

A number of volumes have been added to the Society library during the past year. The Secretary's office is arranging to prepare a list of publications, etc., contained in the library, and this will be printed in an early issue of The Journal.

Files of the A. S. R. E. Journal have been placed in the libraries of the following institutions and publications during the past year:

American Society of Civil Engineers, New York, N. Y.
Franklin Institute, Philadelphia, Pa.
University of Pennsylvania, Philadelphia, Pa.
Columbia University, New York, N. Y.
St. Louis Public Library, St. Louis, Mo.
American Society of Heating and Ventilating Engineers, New York, N. Y.
Bulletin of the Russian Refrigeration Committee, Petrograd, Russia.
Zeitschrift fur Eis und Kalte-Industrie, Vienna, Austria.
Zeitschrift fur die gesamte Kalte Industrie, Munich, Germany.
Armour Institute of Technology, Chicago, Ill.
Ohio State University, Columbus, Ohio.
University of Kansas, Lawrence, Kan.

Our Fourth Western Meeting, in San Francisco, Cal., on September 23 and 24, as a joint meeting with the American Association of Refrigeration, was not very largely attended, although about fifty members and guests were in attendance, and it resulted in interesting refrigerating engineers on the Pacific Coast in the work of the Society, the primary object of holding the meeting in San Francisco. Several applications for membership from refrigerating engineers on the Pacific Coast have been received and others are expected.

On invitation of the officials of the International Engineering Congress, to name some one to preside over its section on refrigeration, your President was pleased to appoint our Vice-President, Theodore O. Vilter, Milwaukee, Wis., to fill this honored position.

The Secretary has been instructed to extend a discount of 10 per cent on subscriptions to the A. S. R. E. JOURNAL received from publishers, subscription agents and book sellers.

Respectfully submitted,

(Signed) LOUIS DOELLING, *President.*

A motion prevailed accepting the report of The Council as submitted.

In referring to the report of the Finance Committee, showing the favorable financial conditions of the Society, President Doelling requested all members who had not paid their dues to try to do so as promptly as possible, that the work of the Society may not be handicapped in any way.

Suggestions followed as to the manner of collecting the dues from the few delinquent members, and on motion of Nelson J. Waite, which was seconded and carried, the Secretary was instructed to enclose with bills to delinquent members a reprint of paragraph C 18 of the Constitution, relating to suspensions and expulsions.

The report of the Tellers of Election of Members was now in order. It is as follows:

REPORT OF TELLERS OF ELECTION OF MEMBERS

NEW YORK, N. Y., December 3, 1915.

Mr. Louis K. Doelling, President,
The American Society of Refrigerating Engineers,
154 Nassau Street, New York, N. Y.

DEAR SIR:

Your Tellers of Election of Members beg to report that they met on the afternoon of December 3, 1915, and canvassed the ballots for new members closing December 2, 1915.

One hundred and thirty-one ballots were cast, five of which were void.

We find that the following have been elected to membership in the grades indicated:

MEMBERS

William I. Bodine, Springfield, Mo.

ASSOCIATE MEMBERS

Alan K. Gillespie, St. Louis, Mo.
Harry Hammond, Alexandria, Va.
Carl Nessler, Texas City, Tex.

JUNIOR MEMBERS

Walter Jones, New Brunswick, N. J.

CHANGE OF GRADE OF MEMBERSHIP FROM JUNIOR MEMBER TO MEMBER

Carl E. Hoffman, Houston, Tex.

Respectfully submitted,
TELLERS OF ELECTION OF MEMBERS
(Signed) W. D. MONKS, *Chairman.*
PETER BINZEL, JR.
JEAN NEÜBECKER.

President Doelling formally accepted the successful applicants as members of the Society in their respective grades.

The reports of committees were next in order, and they were in substance as follows:

Committee on Suggesting a Standard Tonnage Basis of Refrigeration and Standard Method of Testing Refrigerating Machinery, to act jointly with a similar committee of the American Society of Mechanical Engineers—F. E. Matthews, secretary of the committee, reported that Dr. Jacobus, its chairman, had been so occupied with other matters during the past year that he was not able to continue the work of the committee. He further stated that he hoped to be able to make a more favorable report at the next annual meeting.

Committee on Standardization of Ammonia Fittings—The report was made by its chairman, Thomas Shipley. He reported as follows: "As chairman of the Committee on the Standardization of Ammonia Fittings, will say that I can report progress. This apparent lack of interest is the result of an agreement between practically the entire committee, as it is their belief that the time for the standardization of ammonia fittings has not yet arrived."

Committee to Suggest Methods and Promote the Testing of Insulating Materials—In reply to a request for a report from the Committee to Suggest Methods and Promote the Testing of Insulating Materials, its chairman, F. E. Matthews, submitted the following:

The Committee to Suggest Methods and Promote the Testing of Insulating Materials does not feel that it has as yet accomplished anything worthy of the name of a formal report. We prefer, therefore, that what we have to say be considered as discussion.

Your Committee, which it has been suggested might be known simply as the "Insulation Committee," was appointed in 1913 as a result of the general unsatisfactory condition of the technology of the art. There has been a decided lack of accuracy in nomenclature, as well as lack of agreement in the results of experimental work, the latter due probably to diversity of methods employed.

Results of research mean nothing without either a knowledge of all the conditions under which they were obtained or knowledge that they were obtained under some known standard conditions governing other determinations of the same kind, with which results are to be compared. In consideration of the difficulty, and frequently the impossibility, of so interpreting the results of one set of tests as to make them directly comparable with another, it would appear advisable to adopt some standard method for conducting such tests.

The recommendation of such a method, together with a specification of conditions under which the tests should be conducted, is a matter of so great importance that your Committee does not feel that it has as yet given the matter sufficient study to warrant definite recommendations. It hopes, however, to be able to gain the co-operation of the various investigators, so that such methods and conditions may be devised as will be satisfactory to both the experimenter and the engineer, the manufacturer and the consumer.

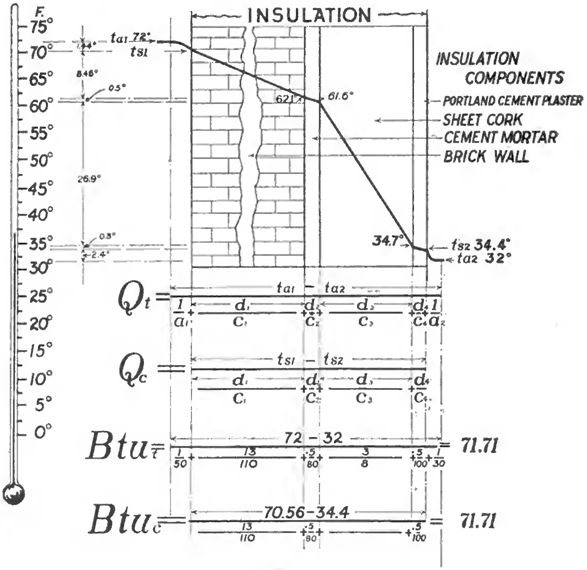
F.
75°
70°
65°
60°
55°
50°
45°
40°
35°
30°
25°
20°
15°
10°
5°
0°
INSULATION
ta1 72°
ts1
1.44°
8.46°
0.5°
26.9°
0.3°
2.4°
62.1°
61.6°
34.7°
ts2 34.4°
ta2 32°
INSULATION COMPONENTS
PORTLAND CEMENT PLASTER
SHEET CORK
CEMENT MORTAR
BRICK WALL
$Q_t = \dfrac{t_{a1} - t_{a2}}{\frac{1}{a_1} + \frac{d_1}{C_1} + \frac{d_2}{C_2} + \frac{d_3}{C_3} + \frac{d_4}{C_4} + \frac{1}{a_2}}$
$Q_c = \dfrac{t_{s1} - t_{s2}}{\frac{d_1}{C_1} + \frac{d_2}{C_2} + \frac{d_3}{C_3} + \frac{d_4}{C_4}}$
$Btu_T = \dfrac{72 - 32}{\frac{1}{50} + \frac{13}{110} + \frac{.5}{80} + \frac{3}{8} + \frac{.5}{100} + \frac{1}{30}} = 71.71$
$Btu_C = \dfrac{70.56 - 34.4}{\frac{13}{110} + \frac{.5}{80} + \frac{.5}{100}} = 71.71$

FIG. 1

It is thought that the question of nomenclature might be cleared up to some extent by considering *insulation* to be the total barrier to heat flow, comprising the aggregate of all the *insulation components* entering into its construction; the *insulation components* comprising all the substances resisting the flow of heat entering into the construction of *insulation*, including those materials against which the insulation component proper is secured, by which it is secured, and with which it is finished, as shown in Fig. 1.

This suggestion may appear revolutionary, but as all building materials are more or less imperfect conductors of heat, there seems to be no suitable place to draw the line between the better conductors and the poorer, which have heretofore been referred to as insulation. Hollow tile, for example, under our present mode of thinking, might or might not be considered an insulating material, although its presence, which we now suggest calling an *insulation component*, imposes considerable resistance to heat flow.

It seems further advisable to emphasize the distinction already made, but not always observed, between the *heat transmission* and heat conduction. The *heat transmission value of insulation* is the total quantity of heat that will pass by *radiation, convection* and *conduction* through a unit area per degree difference in temperature per 24 hours, temperature differences to be taken in the air.

Heat conductivity of an insulation component is the amount of heat that will pass by *conduction only* through a unit area of the insulation component of a unit thickness, per degree difference in temperature per 24 hours, the temperature difference to be taken at the surface of the sample under test.

Your Committee believes the recognition of the applicability of certain established physical laws to the flow of heat by radiation, convection and conduction through *insulation* and *insulation components* used in cold storage work to be fundamental to the work on hand.

For calculating the *heat conduction, Q_c, of insulation* consisting of more than one layer of *insulation components* of known conductivities, the Committee recommends the use of an expression of the form of

$$Q_c = \frac{T_{s_1} - T_{s_2}}{\left(\frac{d_1}{c_1} + \frac{d_2}{c_2} + \frac{d_3}{c_3} \cdots \right)},$$

that is,

$$\left(\begin{matrix}\text{Conducted heat}\\ (Q_c)\end{matrix}\right) = \frac{((T_{s1})\ \text{Temperature of the warmer surface}) - ((T_{s2})\ \text{Temperature of cooler surface})}{\left(\frac{(d_1)\ \text{Thickness of first component}}{(c_1)\ \text{Conductivity of first component}}\right) + \left(\frac{(d_2)\ \text{Thickness of second component}}{(c_2)\ \text{Conductivity of second component}}\right) + \left(\frac{(d_3)\ \text{Thickness of third component}}{(c_3)\ \text{Conductivity of third component}}\right)}$$

from which the heat conduction per degree difference in temperature, per unit area, per unit thickness, per unit of time is

$$\frac{1}{Q_c} = \frac{d_1}{c_1} + \frac{d_2}{c_2} + \frac{d_3}{c_3} \cdots.$$

in which if the *d*'s represent thicknesses in inches of the various samples and *c*'s the respective B. t. u. heat conductions of samples one inch thick, per square foot, per Fahr. degree difference in temperature per twenty-four hours, Q will be expressed in B. t. u. and $\frac{1}{Q_c}$, the reciprocal of the heat conduction, will represent resistance, the measure of effectiveness of the insulation.

The *heat transmission* of the *insulation*, Q_t, which takes place by conduction, convection and radiation, is

$$Q_t = \frac{(T_{a_1} - T_{a_2})}{\left(\frac{1}{a} + \frac{d_1}{c_1} + \frac{d_2}{c_2} + \frac{d_3}{c_3} \dots \frac{1}{a_2}\right)}$$

that is,

$$\left(\begin{matrix}\text{Transmitted heat}\\(Q_t)\end{matrix}\right) = \frac{((T_1)\ \text{Temperature of warm air}) - ((T_2)\ \text{Temperature of cooler air})}{\frac{1}{\left(\begin{matrix}(a_1)\ \text{Heat convection}\\ \text{plus radiation}\end{matrix}\right)} + \left(\frac{(d_1)\ \text{Thickness of first component}}{(c_1)\ \text{Conductivity of first component}}\right) + \left(\frac{(d_2)\ \text{Thickness of second component}}{(c_2)\ \text{Conductiviay of second component}}\right) + \dots \frac{1}{\left(\begin{matrix}(a_2)\ \text{Heat convection}\\ \text{plus radiation}\end{matrix}\right)}}$$

from which the heat transmission per Fahr. degree difference in temperature, per square foot area, per inch of thickness, per 24 hours is

$$\frac{1}{Q_t} = \left(\frac{1}{a_1} + \frac{d_1}{c_1} + \frac{d_2}{c_2} + \frac{d_3}{c_3} \dots + \frac{1}{a_2}\right)$$

The relation between these two equations can be seen by reference to Fig. 1.

Committee to Secure Additional Appropriation from Congress to Enable the Bureau of Standards to Continue Its Investigations of Refrigeration Problems—Theodore O. Vilter, its chairman, reported than an additional appropriation of $15,000 had been secured from Congress in December, 1914.

Committee on Revision of Constitution and By-Laws—The report was made by its chairman, L. Howard Jenks. It is as follows:

New York, N. Y., August 4, 1915.

Mr. Louis Doelling, President,
The American Society of Refrigerating Engineers,
154 Nassau Street,
New York, N. Y.

Dear Sir:

The Revision of Constitution and By-Laws Committee begs to report that it met on the afternoon of August 4, 1915, and considered the suggestions for additions, enlargements and eliminations to the provisions of our Constitution and By-Laws and we submit herewith a revised Constitution and By-Laws.

(Words, paragraphs and sections in italics are new matter, and words and paragraphs in parentheses are to be deleted.)

CONSTITUTION

NAME, OBJECT AND GOVERNMENT

C 1. The title of this Society shall be "The American Society of Refrigerating Engineers."

C 2. The object of this Society is to promote the Arts and Sciences connected with Refrigerating Engineering.

C 3. The principal means for this purpose shall be the holding of meetings for the reading and discussion of appropriate papers, and for social intercourse: the publication and distribution of its papers and discussions; and the maintenance of a library of data on refrigeration.

C 4. The Society shall be governed by this Constitution and by By-Laws and Rules in harmony therewith.

C 5. The Society shall be organized as a Corporation under the Laws of the State of New York.

MEMBERSHIP

C 6. Persons connected with the Arts and Sciences relating to Refrigerating Engineering may be eligible for admission into the Society.

C 7. The membership of the Society shall consist of *Honorary Members,* Members, Associates and Juniors. *Honorary Members,* Members and Associates are entitled to vote and to hold office. Juniors shall not be entitled to vote nor to be officers of the Society, but shall be entitled to the other privileges of membership.

C 8. *Honorary Members,* Members and Associates are entitled to vote on all questions before any meeting of the Society, in person or by proxy, given to a voting member in writing. A proxy shall not be valid for a greater time than six months.

C 9. *An Honorary Member shall be a person of acknowledged professional eminence. The number of Honorary Members shall not exceed five at any time.*

C 10. (9.) A Member shall be twenty-six years of age or over. He must have been so connected with Refrigerating Engineering as to be competent as a designer or as a constructor, to take responsible charge of work in his branch of Refrigerating Engineering, or he must have served as a teacher of Refrigerating Engineering for more than five years.

C 11. (C. 10.) An Associate shall be twenty-six years of age or over and shall be so connected with Refrigerating Engineering as to be competent to take charge of engineering work, or to co-operate with Refrigerating Engineers.

C 12. (C 11.) A Junior shall be twenty-one years of age or over. He must have had such Refrigerating Engineering experience as will enable him to fill a responsible subordinate position in Refrigerating Engineering work, or he must be a graduate of an engineering school.

C 13. (C 12.) The rights and privileges of every *Honorary Member,* Member, Associate and Junior shall be personal to himself, and shall not be transferable or transmissible by his own act or by operation of law.

ADMISSION

C 14. Honorary Members shall be nominated by at least ten Members of the Society. The grounds upon which the nomination is made shall be presented to The Council in writing.

C 15. (C 13.) All applications for membership as Member, Associate or Junior shall be presented to The Council, which shall consider and act upon each application, assigning each approved applicant to the classification to which, in the judgment of The Council, he is entitled. The name of each candidate thus approved by The Council shall, unless objection is made by the applicant, be submitted to the voting membership for election by means of a letter ballot.

C 16. (C 14.) Associates or Juniors desiring to change their grade of membership shall make application to The Council in the same manner as is required in the case of a new applicant.

C 17. (C 15.) Election to membership shall be by a sealed letter ballot as the By-Laws shall provide. Adverse votes to the number of four per cent. of the votes cast shall be required to defeat the election of an applicant.

C 18. (C 16.) Each person elected shall subscribe to this Constitution, and shall pay the initiation fee before he can be entitled to the rights and privileges of membership. If such person does not comply with these requirements within six months after notice of his election, he will be deemed to have declined election. The Council may, thereupon, declare his election void.

INITIATION FEES AND DUES

C 19. (C 17.) The initiation fee for membership shall be as follows:

For Members and Associates, $5.

For Juniors, $5. A Junior on promotion to any other grade of membership shall pay an additional fee of $5.

The annual dues for membership shall be as follows, *including $3 for a subscription to the A. S. R. E. Journal for one year:*

For Members and Associates *residing in the United States and Canada,* $15; *elsewhere, $10.*

For Juniors *residing in the United States and Canada,* $10; *elsewhere, $5* (for the first six years of their membership, and thereafter the same as for a Member or Associate).

C 20. The Council may, at its discretion, permit any Member or Associate to become a Life Member in the same grade, without further payment of dues, by the payment in a single sum of an amount corresponding to the value of an annuity equal to the annual dues at the age of the applicant at time of payment, according to "McClintock's Table of Mortality among Annuitants with Interest at 3½ per cent per Annum," which table is the basis of valuation for annuities as prescribed by the State of New York.

SUSPENSIONS AND EXPULSIONS

C 21. (C 18.) Any Member, Associate or Junior who shall leave his annual dues unpaid for two years shall, at the discretion of The Council, have his name stricken from the roll of membership and shall cease to have any further rights of membership.

C 22. (C 19.) The Council may refuse to receive the dues of any Member, Associate or Junior who shall have been adjudged by The Council to have violated the Constitution or By-Laws of the Society, or who, in the opinion of The Council, expressed by a two-thirds vote of the entire Council, shall have been guilty of conduct rendering him unfit to continue in its membership; and The Council may expel such person and remove his name from the list of members.

THE COUNCIL

C 23. (C 20.) The affairs of the Society shall be managed by a Board of Directors chosen from among its Members and Associates, which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, the two Vice-Presidents, Treasurer, (and) fifteen Members and Associates, *and the last five living Ex-Presidents, ex-officio.* Seven members of The Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of The Council, but shall not have a vote therein.

C 24. (C 21.) The Council thus constituted shall regulate its own proceedings and shall be the legal Trustee of the Society. All gifts or bequests not designated for a specific purpose shall be invested by The Council and only the income therefrom may be used for current expenses.

C 25. (C 22.) Should a vacancy occur in The Council or in any elective office except the presidency, through death, resignation or other cause, The Council may elect (a) *an Honorary Member,* Member or Associate to fill the vacancy until the next annual election.

C 26. (C 23.) The Council shall present at the Annual Meeting of the Society a report, verified by the President or Treasurer or by a majority of the members of The Council, showing the whole amount of real and personal property owned by the Society, where located and where and how invested, and the amount and nature of the property acquired during the year immediately preceding the date of the report, and the manner of the acquisition; the amount applied, appropriated or expended during the year immediately preceding such date, and the purpose, objects or persons to or for which such applications, appropriations or expenditures have been made: also the names and places of residence of the persons who have been admitted to membership in the Society during the last year, which report shall be filed with the records of the Society, and an abstract thereof shall be entered in the minutes of the proceedings of the Annual Meeting.

C 27. (C 24.) An act of The Council which shall have received the expressed or the implied sanction of the membership at the next subsequent meeting of the Society shall be deemed to be the act of the Society, and shall not afterward be impeached by any member.

C 28. (C 25.) The Council may, by two-thirds vote of the members present, declare any elective office vacant, on the failure of its incumbent for one year, from inability or otherwise, to attend The Council meetings, or to perform the duties of his office, and shall thereupon appoint (a) *an Honorary Member,* Member or Associate to fill the vacancy until the next Annual Meeting. The said appointment shall not render the appointee ineligible to election to any office.

OFFICERS

C 29. (C 26.) At each Annual Meeting there shall be elected from among the *Honorary Members,* Members and Associates:

A President to hold office for one year.

Two Vice-Presidents, one to hold office for one year and one to hold office for two years. After the first year one Vice-President to be elected annually for a term of two years.

A Treasurer to hold office for one year.

Fifteen Members or Associates shall be members of The Council, five being elected each year, and each to hold office for three years, with the exceptions contained in this paragraph. On the year of acceptance of this amendment (1912), in order to increase the number of Directors from nine, as heretofore, to fifteen, as provided in this paragraph and in amended Section C 20, nine Members or Associates shall be elected to The Council, five to hold office for three years, two to hold office for two years, and two to hold office for one year, thereafter five Members or Associates shall be elected each year.

C 30. (C 27.) The election of officers shall be by ballot, as the By-Laws shall provide.

C 31. (C 28.) The term of all elective officers shall begin on the adjournment of the Annual Meeting of the Society. Officers shall continue in their respective offices until their successors have been installed.

C 32. (C 29.) A President or Vice-President shall not be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

C 33. (C. 30.) The Council, at its first meeting after the Annual Meeting of the Society, shall appoint a Member or Associate to serve as Secretary of the Society, subject to the pleasure of The Council. The Secretary shall receive a salary which shall be fixed by The Council at the time of his appointment.

C 34. (C 31.) The President, Secretary and Treasurer shall perform the duties legally or customarily attaching to their respective offices under the laws of the State of New York, and such other duties as may be required of them by The Council.

C 35. (C 32.) A vacancy in the office of President shall be filled by the Vice-President who is senior by age.

MEETINGS

C 36. (C 33.) The Society shall hold its Annual Meeting in New York City on the Monday preceding the first Tuesday in December, and such other meeting shall be held at such times and places as The Council may appoint.

Twenty-five Members and Associates shall constitute a quorum for the transaction of business.

C 37. (C 34.) Special meetings of the Society may be called at any time at the discretion of The Council or shall be called by the President upon the written request of twenty-five members entitled to vote.

C 38. (C 35.) Any appropriation recommended by the Society at a meeting shall not take effect until it has been approved by The Council.

C 39. (C 36.) Every question which shall come before a meeting of the Society or of The Council or of a committee shall be decided by a majority of the votes cast, unless otherwise provided in this constitution or by the By-Laws, or the Laws of the State of New York. The Council may order the submission of any question to the membership for discussion by letter ballot. Any meeting of the Society at which a quorum is present may order the submission of any question to the membership for discussion by letter ballot.

STANDING COMMITTEES

C 40. (C 37) The standing committees of the Society shall be:

Finance Committee.
Publication Committee.
Membership Committee.

The members of these committees shall be appointed by the President from members of The Council who are not officers of the Society.

TRANSACTIONS

C 41. (C 38.) The Society shall not be responsible for statements or opinions advanced in papers or in discussions at its meetings. Matters relating to politics, religion or purely to trade shall not be discussed at a meeting of the Society nor be included in the Transactions.

C 42. (C 39.) The Society shall not approve any engineering or commercial enterprise, nor allow its imprint or name to be used in any commercial work or business. No Member shall describe himself in connection with the Society in any advertisement other than as (a) *an Honorary Member,* Member, Associate Member or Junior Member.

C 43. Branch Associations of The American Society of Refrigerating Engineers may be established by the adoption of a Constitution, which shall be approved by The Council of the parent Society, and by organization of the proposed Branch Association thereunder, when, at the discretion of The Council, a sufficient number of members makes application to that effect.

The Constitution of Branch Associations shall provide:

a. That all members of Branch Associations shall be members, in good standing, of The American Society of Refrigerating Engineers.

b. That all fees and dues assessed by the Branch Associations against their members shall be in addition to regular fees and dues required by and paid to the parent Society, except that The Council of the parent Society may appropriate not more than $3 of the dues of each Member and Associate and not more than $2 for each Junior, who are members of the Branch Associations, to be applied to the expenses of the Branch Associations.

A copy of the minutes and proceedings of each meeting of Branch Associations shall be filed with the Secretary of The American Society of Refrigerating Engineers within fifteen days after the meeting. The American Society of Refrigerating Engineers may publish in the A. S. R. E. Journal such proceedings of Branch Associations as shall be approved by the Publication Committee, but no part of the proceedings shall be published without the approval of the Publication Committee of the parent body.

AMENDMENTS

C 44. (C 40.) At Annual Meetings of the Society any *Honorary Member,* Member or Associate may propose in writing for discussion an amendment to this Constitution. Such proposed amendment shall not be voted on at that meeting, but shall be open for discussion and such modifications as may be accepted by the proposer. The proposed amendment shall be mailed by the Secretary to each *Honorary Member,* Member and Associate at the time the notice of the next Annual Meeting issues, and shall be voted upon at said meeting.

C 45. (C 41.) Such By-Laws shall be enacted as will conform with this Constitution and the Laws of the State of New York and as are required to conduct the business of the Society.

BY-LAWS

CANDIDATES FOR MEMBERSHIP

B 1. A candidate for admission to the Socety as a Member or as an Associate must make application on a form approved by The Council, upon which he shall write a statement giving a complete account of his qualifications and engineering experience, and an agreement that he will, if elected, conform to the Constitution, By-Laws and Rules of the Society. He must refer to at least four Members or Associates to whom he is personally known.

B 2. Applications for membership from Refrigerating Engineers who are not residents in the United States or Canada, and who may be so situated as not to be personally known to four Members or Associates of the Society, as required in the foregoing paragraph, may be recommended for ballot by four members of The Council, after sufficient evidence has been secured to show that in their opinion the applicant is worthy of admission to the grade which he seeks.

B 3. A candidate for admission to the Society as a Junior must make application in the same manner as provided for Members or Associates, except that he must refer to not less than three Members or Associates to whom he is personally known.

B 4. The references for each candidate for admission to the Society shall be requested to make a confidiential communication to the Membership Committee, setting forth in detail such information, personally known to the referee, as shall enable The Council to arrive at a proper estimate of the eligibility of the candidate for admission to the Society.

ELECTION OF MEMBERS

B 5. The Secretary shall mail to each member entitled to vote, at least thirty days in advance of any meeting, a ballot stating the names and the respective grades of the candidates for membership in the Society which have been approved by The Council and the time of the closure of voting. The voter shall prepare his ballot by crossing out the names of candidates rejected by him, and shall enclose said ballot in a sealed blank ballot envelope, which he shall then enclose in a second sealed outer envelope, on which he shall, for identification, write his name in ink. The ballot thus prepared and enclosed shall be mailed or delivered unopened to the Tellers of Election. The Secretary shall certify to the competency and the signature of all voters. On the closure of voting the Tellers of Election shall first open and destroy the outer envelopes, and shall then canvass the ballots and certify the result to the meeting of the Society.

B 6. The Tellers of Election shall not receive any ballot after the stated time of the closure of voting. A ballot without the endorsement of the voter written in ink on the outer envelope is defective, and shall be rejected by the Tellers of Election.

B 7. The names of those persons elected to membership, with their respective grades, shall be embodied in a written report, signed by the Tellers, and presented to the next meeting of the Society. The President shall then declare them duly elected to membership in the Society. The Tellers may, through the Secretary, in advance of any meeting, advise each candidate of the result of the canvass of the votes in his case. The names of applicants who are not elected shall neither be announced nor recorded in the Transactions.

B 8. The endorsers of an applicant who has not been elected may, with his consent, present to The Council a written request for a re-submission of his name to ballot. The Council may, in its discretion, by a three-fourths vote of the members present order the name of the applicant placed on the next ballot for members.

B 9. 'Each person elected to membership must subscribe to the Constitution, By-Laws and Rules of the Society, and pay the initiation fee before he can receive a certificate of membership in the Society.

ELECTION OF OFFICERS

B 10. The Secretary shall mail to each member entitled to vote, at least thirty days before the Annual Meeting, the names of the candidates for office proposed for election by the Nominating Committees.

B 11. The names of the candidates proposed by the Nominating Committee or Committees, and the respective offices for which they are candidates, shall be printed in separate lists on the same ballot sheet, each list of candidates to be printed under the names of the members of the particular committee which proposed it.

B 12. The name of any candidate on the ballot may be erased, and the name of any person qualified to hold the office written in its stead. The ballot must be voted and canvassed in the same manner as for the election of members.

B 13. In case of a tie in the vote for any office, the President, or in his absence, the presiding officer, shall cast the deciding vote.

B 14. A ballot which contains more names on it than there are officers to be elected is thereby defective, and shall be rejected by the Tellers.

FEES AND DUES

B 15. The initiation fee and annual dues of the first year shall be due and payable on notice of election to membership. Thereafter the annual dues shall be due and payable on the first day of December in each year.

B 16. A member in arrears for one year shall not be entitled to vote until such arrears have been paid. Should the right to vote be questioned, the books of the Society shall be conclusive evidence.

B 17. The Secretary shall present to The Council the name of any Member, Associate or Junior in arrears for more than one year. A person dropped from the rolls for non-payment of dues may, at the discretion of The Council, be restored to the privileges of membership upon payment of all arrears.

FINANCIAL ADMINISTRATION

B 18. The Council at its first meeting in each fiscal year shall consider the recommendations of the Finance Committee concerning the expenditures necessary for the work of the Society during that year. The apportioning of the work of the Society among the various standing and other committees shall be on a basis approved by The Council and in harmony with the Constitution and By-Laws. The appropriations approved by The Council, or so much thereof as may be required for the work of the Society, shall be expended by the various committees of the Society, and all bills against the Society for such expenditures shall be certified by the committee making the expenditure, and shall then be sent to the Finance Committee for audit. Money shall not be paid out by any officer or employee of the Society except upon bills duly audited by the committee, or by resolution of The Council.

FINANCE COMMITTEE

B 19. The Finance Committee shall consist of three members of The Council appointed for a term of one year. The committee shall, under the direction of The Council, have supervision of the financial affairs of the Society, including the books of account.

PROGRAM COMMITTEE

B 20. The President shall appoint a Program Committee from the members of The Council, which shall procure professional papers, pass upon their suitability for presentation, and to suggest topical subjects for discussion at the meetings. The committee may refer any paper presented to the Society to a person or persons, especially qualified by theoretical knowledge or practical experience, for their suggestions or opinions as to the suitability of the paper for presentation. Papers from non-members shall not be accepted except by unanimous vote of the committee. The committee shall arrange the program of each meeting of the Society, and shall have general charge of the entertainments to be provided for the members and guests at each meeting. It shall prohibit the distribution or exhibition at the headquarters or at the meeting places of the Society of all advertising circulars, pamphlets or samples of commercial apparatus or machinery. At the end of each fiscal year the committee shall deliver to the Secretary for presentation to The Council a detailed report of its work.

PUBLICATION COMMITTEE

B 21. The Publication Committee shall consist of three members of The Council appointed for a term of one year. The committee shall review all papers and discussions which have been presented at the meetings, and shall decide what papers or discussions, or parts of the same, shall be printed. At the end of each fiscal year the committee shall deliver to the Secretary for presentation to The Council a detailed report of its work.

MEMBERSHIP COMMITTEE

B 22. The Membership Committee shall consist of three members of The Council appointed for a term of one year. It shall be the duty of this committee:

To receive and scrutinize all applications for membership to the Society.

To seek further information as to the qualifications of any applicant whose evidence of eligibility is not clear to the committee.

To report to each session of The Council the names of all applicants under consideration, together with the action of the committee on each.

The committee shall at once destroy all correspondence in relation to each applicant when his name has been placed on the ballot by order of The Council, or upon the withdrawal of the application.

NOMINATING COMMITTEES

B 23. A Nominating Committee of five Members or Associates shall be appointed by the President within three months after he assumes office. It shall be the duty of this committee to send to the Secretary on or before October first the names of consenting nominees for the elective offices next falling vacant under the Constitution. Upon the request of any *Honorary Member,* Member or Associate, the Secretary shall furnish to the applicant the names of such nominees.

B 24. Twenty or more members entitled to vote may constitute themselves a Special Nominating Committee with the same powers as the Annual Nominating Committee. A Special Nominating Committee, if organized, shall, on or before October twentieth, present to the Secretary the names of the candidates nominated by it for the elective offices next falling vacant under the Constitution, together with the written consent of each.

TELLERS

B 25. The Presiding Officer shall, at the first session of the Annual Meeting, appoint three Tellers of Election of Officers, whose duties shall be to canvass the votes cast, and report the result to the meeting. Their term of office shall terminate when their report of the canvass is presented to the meeting.

B 26. The President within one month after assuming office shall appoint three Tellers of Election of Members to serve for one year, whose duties shall be to canvass the votes cast for members during the year, and to certify the same to the President. They shall notify candidates through the Secretary of the result of such election.

B 27. The President shall appoint three Tellers to canvass any letter ballots which shall be order by The Council or by the Society.

SECRETARY

B 28. The Secretary of the Society shall be the Secretary to The Council.

The Secretary shall, under the supervision of the Finance Committee, have charge of the Books of Account of the Society.

He shall make and collect all bills against members or others.

All bills against the Society shall be delivered to the Secretary. He shall immediately enter them in the Books of Account, and shall immediately deposit such funds as he receives to the credit of the Society in a bank to be designated by The Council.

TREASURER

B 29. The Treasurer shall make payments only on the audit of the Finance Committee, or upon the direction of The Council, by resolution of that body. He shall furnish a bond for the faithful performance of his duties to such amount as The Council may require, such bond to be procured from an incorporated guarantee company at the expense of the Society.

TITLES, EMBLEMS, CERTIFICATES

B 30. Each *Honorary Member,* Member, Associate and Junior, subject to such rules as The Council may establish, shall be entitled on request to a certificate of membership signed by the President and Secretary of the Society. Every such certificate shall remain the property of the Society, and shall be returned to it on demand of The Council.

B 31. Each proxy authorizing a person to vote for an absent member shall be signed by such absent member, with an attesting witness, and be submitted to the Secretary for verification of the member's right to vote at the meeting at which the right is to be exercised.

B 32. The emblem of each grade of membership approved by The Council shall be worn by those only who belong to that grade. The official stationery shall be used only by Officers and Committees of the Society for official business.

B 33. The abbreviations of the titles of the various grades of membership approved by the Society are as follows:

For Honorary Member—Hon. Mem. Am. Soc. R. E.
For Members—Mem. Am. Soc. R. E.
For Associates—Assoc. Am. Soc. R. E.
For Juniors—Jun. Am. Soc. R. E.

ORDER OF BUSINESS

B 34. Roll Call.
Reading of minutes of previous meeting.
Report of Tellers of Election of Members.
Report of The Council.
Unfinished business.
New business.
Report of Tellers of Election of Officers.
Reading of papers and discussions.

PARLIAMENTARY RULES

B 35. In all questions arising at any meeting involving parliamentary rules not provided for in these By-Laws, "Cushing's Manual" shall be the governing authority.

AMENDMENTS

B 36. These By-Laws may be amended in the same manner as amendments are made to the Constitution.

SUGGESTION FOR CONSTITUTION OF BRANCH ASSOCIATION OF THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

Your Committee also submits the following Constitution for Branch Associations:

ARTICLE I

NAME, LOCATION AND OBJECT

Section 1. The name of the Association shall be the .... Association of Members of The American Society of Refrigerating Engineers.

Sec. 2. The officers of the Association shall be located in the city of ....

Sec. 3. Its object shall be the advancement of refrigerating engineering knowledge and practice, the maintenance of a high professional standing among its members, and the furtherance of the general welfare of The American Society of Refrigerating Engineers.

Sec. 4. Among the means to be employed for this purpose shall be the holding of meetings for the presentation and discussion of professional papers, particularly those issued by The American Society of Refrigerating Engineers, and the encouragement of engineers who are not members to attend and take part in the professional and social features of such meetings.

ARTICLE II

MEMBERSHIP

Section 1. The membership of the Association shall be restricted to persons who are members of The American Society of Refrigerating Engineers in any grade. All such may become members of this Association without payment of entrance fee, but should the membership in The American Society of Refrigerating Engineers of any member cease from any cause he shall at the same time cease to be a member of this Association.

ARTICLE III

DUES

Section 1. The annual dues payable by all members shall not be more than ...., payable annually in advance on the first day of ...., and any member whose dues are more than two years in arrears shall cease to be a member of this Association.

ARTICLE IV

OFFICERS

Section 1. The officers of this Association shall be a President, two Vice-Presidents, a Secretary and a Treasurer, who, with the two latest living past-presidents, shall constitute a Board of Directors, in which the government of the Association shall be vested.

Sec. 2. The terms of office of the President of the Association shall be one year, of the Vice-Presidents, Secretary and Treasurer, two years. The term of each officer shall begin at the close of the annual meeting at which such officer is elected, and shall continue for the period above named, or until a successor is duly elected.

Sec. 3. A vacancy in any office shall be filled by election at the next regular meeting of the Association.

Sec. 4. The officers following those elected at the organization of the Association shall be elected at the annual meetings of the Association thereafter.

ARTICLE V

MEETINGS

Section 1. The annual meeting of the Association shall be held on...... ....

Sec. 2. Regular meetings shall be held bi-monthly, at such time and place as may be appointed by the Board of Directors. At these meetings the business of the Association shall be transacted, and the discussion of professional papers published by The American Society of Refrigerating Engineers, or specially arranged for the occasion, shall be in order. Such discussions are to be furnished to the Secretary of The American Society of Refrigerating Engineers for publication, subject to the approval of the Publication Committee of that Society.

ARTICLE VI

AMENDMENTS

Section 1. This Constitution may be amended only by a two-thirds vote of all the members of the Association, which vote may be taken, if necessary, by letter ballot, provided such amendment shall have previously received the approval of The Council of The American Society of Refrigerating Engineers.

Your Committee approves the amendments to Paragraph C 17 and C 20, offered by Ex-President Torrance at our last annual meeting, as to readjustment of dues for foreign members and the making of an addition to The Council of the last five living ex-presidents, and have included these amendments in our suggestions,

Respectfully submitted,

L. HOWARD JENKS, *Chairman.*

The report was ordered printed in The Journal and then brought before the next annual meeting for final action. However, Chairman Jenks urged the acceptance of the amendment to paragraph 4 of Section C 17 of the Constitution, stating that the postal authorities require that this amendment be made before they will admit the A. S. R. E. JOURNAL to the privileges of second class mail matter. Paragraph 4 of Section C 17 reads as follows:

"The annual dues for membership shall be as follows":

Mr. Jenks stated that the postal authorities desire that it read as follows:

"The annual dues for membership shall be as follows, including $3.00 for a subscription to the 'A. S. R. E. Journal' for one year":

A motion prevailed accepting this amendment to paragraph 4 of Section C 17 of the Constitution.

At the last annual meeting the President, Henry Torrance, on recommendation of The Council, proposed the following amendments to paragraphs C 17 and C 20 of the Constitution, subject to discussion and such modifications as may be accepted by the proposer at that meeting and to come before the Eleventh Annual Meeting for final action.

(Words in italics indicate new matter; words in parentheses to be omitted.)

INITIATION FEES AND DUES

C 17. The initiation fee for membership shall be as follows:

For Members and Associates, $5.

For Juniors, $5. A Junior on promotion to any other grade of membership shall pay an additional fee of $5.

The annual dues for membership shall be as follows:

For Members and Associates *residing in the United States and Canada*, $15; *elsewhere, $10.*

For Juniors *residing in the United States and Canada*, $10; *elsewhere, $5* (for the first six years of their membership, and thereafter the same as for a Member or Associate).

THE COUNCIL

C 20. The affairs of the Society shall be managed by a Board of Directors chosen from among its Members and Associates, which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, the two Vice-Presidents, Treasurer, (and) fifteen Members and Associates, *and the last five living Ex-Presidents, ex-officio*. Seven members of The Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of The Council, but shall not have a vote therein.

These proposed amendments were unanimously approved.

E. F. Miller, Boston, Mass., suggested that, as a great many of the technical schools are now giving courses in refrigeration and that students naturally desire to secure copies of the A. S. R. E. JOURNAL, a special subscription rate be made to students. This matter was referred to The Council with power to act.

The report of the Tellers of Election of Officers was next in order. It is as follows:

NEW YORK, N. Y., December 6, 1915.

Mr. Louis K. Doelling, President,
The American Society of Refrigerating Engineers,
154 Nassau Street, New York, N. Y.

DEAR SIR:

Your Tellers of Election of Officers met this morning and canvassed the ballots for the nominations for the offices falling vacant after this annual meeting. We find the following were elected:

PRESIDENT

Theodore O. Vilter, Milwaukee, Wis.

VICE-PRESIDENT

N. H. Hiller, Carbondale, Pa.

TREASURER

George A. Horne, New York, N. Y.

DIRECTORS (5 YEARS)

G. F. Bein, Fort Worth, Tex.
Henning N. Borgstedt, Yonkers, N. Y.
Richard S. Broas, New York, N. Y.
Herbert S. Nulsen, New York, N. Y.
George E. Wells, St. Louis, Mo.

DIRECTOR (1 YEAR)

S. B. Carpender, New Brunswick, N. J.

Respectfully submitted,
(Signed) JOHN R. LIVEZEY, *Chairman*.
DONALD COLE.
NELSON J. WAITE.

President Doelling appointed Ex-Presidents Neff and Block to escort President-elect Vilter to the chair. On reaching the rostrum, President Doelling congratulated the President-elect and assured him that he knew he would carry on the business of the Society to the satisfaction of all.

President-elect Vilter made the following remarks in accepting the presidency of the Society:

"I had no idea, gentlemen, that such an honor would come to me. I want to thank you for it. I shall try to do the very best I can, and in a year from now you will be able to judge as to whether I was entitled to have this office or not. I also want your assistance, because one man sitting in this chair here can't do much of anything, he has to have some standbys, and I want those standbys to come forward and say what ought to be done, and should be done and must be done, and we will try and get our heads together and do it, but we must have co-operation. If we haven't co-operation our individual efforts are of little avail, but we certainly want a live organization, and with your co-operation we will make it so.

Cold Storage Legislation Committee—In the absence of its chairman, Henry Torrance, President Doelling called on J. F. Nickerson for a statement on the present situation as to cold storage legislation. Mr. Nickerson stated that there was little new to report since the last meeting, as Congress and none of the state legislatures had been in session for some time. However, he felt that as eleven state legislatures will be in session during 1916, there will be, no doubt, efforts made to introduce and pass cold storage bills. He further stated that it was possible that the bill which failed of passage in the last Congress may be again introduced. He also called attention to the cold storage legislation situation in Pennsylvania, remarking that a bill had passed both houses at the last session of the state legislature repealing the present law and substituting a new law very similar to the uniform state cold storage law, but it was vetoed by the Governor.

C. H. Brownell, Washington Court House, Ohio, in discussing the cold storage law situation, stated that it was his opinion that cold storage laws would have no beneficial effect upon the cold storage warehousemen and the public, for the reason that it is impossible to enforce any law that would compel the retailer to deal honestly with his customers, so long as there was a profit in taking advantage of the public prejudice against cold storage goods, as is now being generally done. He stated that in Massachusetts, where cold storage goods are supposed to be marked, that inferior cold storage eggs or inferior fresh eggs were being stamped as cold storage products, while first class cold storage products were sold as fresh goods. He firmly believed, because of human frailties, that whenever there is an opportunity for an excess profit to the retailer he will take the profit and not conform to the law.

Before adjourning the session for luncheon in the Engineers Club, President Doelling appointed a Resolutions Committee, consisting of J. F. Nickerson, Chicago, Ill.; F. E. Matthews, Leonia, N. J., and Karl Vesterdahl, New York, N. Y.

Second Session, Monday Afternoon, December 6

Immediately upon calling the second session of the meeting to order at about 1:30 o'clock P. M., President Doelling asked for the report of the Committee to Determine the Sources of Foreign Gases in Refrigerating Systems. The report was made by H. Dannenbaum, the chairman of the committee. It is as follows:

The Committee to Determine the Sources of Foreign Gases in Refrigerating Systems found it difficult to continue its work along the lines pursued last year, and considered it best to secure the cooperation of the Bureau of Standards at Washington, D. C., on its important work. An arrangement was made under which Mr. McKelvy of the Bureau, who was placed in charge of this investigation, would take regularly samples of non-condensible gases from a plant at Washington, analyzing them at the laboratory of the Bureau. We visited in the spring several refrigerating plants in Washington and arranged with one of the larger plants, operating compression machines, to take samples whenever they thought the system needed purging. Mr. McKelvy did not hear from the plant for a long time and thought this was due to the prevailing cool weather, which might have an influence on the formation of non-condensible gases, but when he finally visited the plant he was told that it would cause too much trouble and disturbance to have samples of gases taken from the system.

Mr. McKelvy came to the conclusion that it would be a better plan to install at the Bureau of Standards a half-ton compression refrigerating plant for the purpose of thoroughly studying the problem under his absolute control and made an effort to obtain such a plant, but, as the funds of the Bureau are limited, the Director did not see his way clear to let him have the necessary amount of money. An effort was made to have a plant loaned for a year or two, but so far he has not been successful. Mr. McKelvy hopes to obtain a small plant next year and the subject will then be studied on a scientific basis. An article by Prof. R. Planck, Danzig, Germany, which appeared in the *Zeitschrift fuer die gesamte Kaelte Industrie*, a translation of which appeared in the July number of the A. S. R. E. JOURNAL, offers some valuable suggestions for the investigations.

A separate study is the formation of non-condensible gases in absorption machines and some figures were obtained at a plant in Washington operating such machines. A report by Mr. McKelvy on the work which has been done this year follows:

The experimental work on the subject of foreign gas formation in ammonia refrigerating systems undertaken by the Bureau of Standards is not sufficiently advanced to permit of even a preliminary survey of results at this time. However, it was thought that it might be of some interest to point out the lines of attack now being followed and the possible needs of the problem in order to provide definitive and authoritative results.

The two types of systems, absorption and compression, are being considered independently in relation to the problem, as the conditions and factors involved are for the most part different. The results so far obtained have to do with the absorption system. While not conclusive or extensive in nature they are sufficient to indicate the line of attack to be followed. Work is under way along two distinct lines:

1. During a recent overhauling and recharge of the Bureau absorption machine, outlets were provided by means of which it will be possible to take gas samples for analysis from various parts of the machine independently,

thus enabling us to lecoalize the place of formation, study the behavior of the gas in the system and the variations in its composition. Of considerable importance will be the study of the actual quantity of gas formed and its effect on operating conditions. In order to do this, it will be necessary to operate under a load as constant as can be maintained.

Preliminary experiments at the Terminal Power Company plant in Washington, with this object in view, were made during July. The results obtained were only qualitative in character, owing to the variation in load at the plant. During this month 1.5 cubic feet of foreign gas mixed with 14.3 pounds (302 cubic feet) of ammonia were removed from an 80-ton machine. The average composition by volume of the gas blown off was about 0.5 per cent foreign gas, the remainder being ammonia. Here, it is evident, purging was unnecessary. From a 50-ton machine at the same plant 3.3 cubic feet of foreign gas mixed with 2.5 pounds (53 cubic feet) of ammonia was removed. Thus the average composition by volume of the gas blown off was 6.2 per cent foreign gas, the remainder being ammonia.

At the Bureau plant it is hoped to be able to definitely determine the relation between operating conditions and gas formation, as well as the effect of purging on operating conditions.

2. An apparatus has been built in which it is hoped to study the formation of foreign gas on a small scale. The conditions obtaining in an absorption plant will be imitated as closely as possible, and more particularly the effect of impurities on the ammonia will be studied. It is also contemplated using various brands of ammonia on the market. It will be possible to control the various factors involved very closely.

In order to throw light on the behavior of foreign gas in the system, a study of the solubility of the component gases in liquid ammonia at various temperatures and pressures is being made. In connection with this problem, analyses have been made of the gases that frequently occur in hot water heating systems. Their composition shows a surprising similarity to the composition of the foreign gas usually found in absorption systems.

It is expected to use the apparatus that has been devised for imitating conditions in the absorption plant for the same purpose with regard to the compression system. It is realized that it will be much more difficult to obtain the exact conditions in this type of system, and it is hoped that it will be possible to install a small unit compressor for this purpose. All the factors involved in the gas formation will be carefully studied in an experimental way, in particular the relation of impurities in the ammonia and the nature of the oil used to this formation.

The interest that has developed in regard to gas formation in refrigerating systems is evidenced by recent articles in *Zeitschrift fuer gesamte Kaelte Industrie* appearing in Germany. Chief among these is an article by Prof. R. Planck, treating the decomposition of ammonia from the standpoint of chemical dynamics. Previous work on the subject is critically reviewed, and though no new experimental data are presented, it is a very valuable contribution to the literature of the subject. The various factors that influence the rate of decomposition are discussed and their relative significance made clear. Recommendations as to operation in order to avoid decomposition of ammonia are then presented. Several important considerations have been overlooked in the author's consideration of the subject; for example, the possible influence of moisture introduced into the system, of other impurities in the ammonia, and the relation of the oil, always present, to the gas formed. The author's theoretical considerations demand that the composition of the gas formed be 75 per cent hydrogen and 25 per cent nitrogen by volume; that is, the stoichiometrical relations of nitrogen and hydrogen in ammonia. As a matter of fact, in gases that have been analyzed the nitrogen generally exists in a much higher percentage. While some of this loss of hydrogen may possibly be due to faster leakage, it is probable that other factors are at work in causing gas formation. In one case, however, the amount of hydrogen was much greater than it should be, if there was merely a straight decomposition of ammonia.

The results of this theoretical paper emphasize the need of a careful experimental study of the problem, such as is under way at the Bureau of Standards.

Referring to Mr. McKelvy's report on the findings at the absorption plant in Washington, I understand that the samples of gas were taken when it was thought to be necessary to purge the plant. It is quite evident that, with only .5 per cent of non-condensible gas, purging would not be necessary. The smaller machine had a higher percentage of non-condensible gas, but this may be due to the fact that the absorbers were operated under different conditions. Mr. McKelvy refers in his report to analyses of gases which frequently occur in hot water heating systems. The samples of the gases were taken from hot water radiators in his residence, and it is interesting to learn that their composition shows similarity to composition of non-condensible gases in absorption machines. Mr. McKelvy told me that the gases consisted of hydrogen and nitrogen.

In thanking Mr. McKelvy for his kind preliminary report, the committee wishes to express the hope that the Bureau of Standards will soon get the small compression machine, so that by a year from now it will be able to present actual facts on this important problem. If the Society can be of any help to the Bureau in obtaining such a plant, it certainly would be very desirable.

President Doelling remarked about the important work the committee had before it and felt the refrigerating industry should provide the Bureau of Standards with a half-ton refrigerating system, stating that he was willing to supply the compressor. Theodore O. Vilter promptly agreed to supply the condenser and receiver; Henry Torrance the expansion piping and valve, and J. S. Louis, Cincinnati, Ohio, the motor.

President Doelling now made the following address:

ADDRESS BY PRESIDENT DOELLING

The beginning of my address must be dated one year back, to the Tenth Annual Meeting in 1914. Through circumstances not under my control I was unable to be present at that time, and, therefore, it is only to-day that I can thank you personally for the honor you bestowed upon me in making me President of your Society. In electing me you took great chances, because there was a possibility that I might never have reached the United States during my term of office, and, moreover, you elected for your President a man absolutely inexperienced in conducting the affairs of an engineering society. That I am here is due to the fact that I, too, took chances in returning to the United States, and that I was able to perform my duties as President I owe to the cheerful cooperation of The Council, the different committees, and last, but not least, of our able and experienced Secretary, Mr. Ross.

The development of the Society during the last year has been satisfactory. The number of members has been increased, although the Membership Committee has been very careful in its recommendations for admission of new members. The finances are in good condition, and the value and circulation of The Journal are increasing, and it is being recognized as one of the leading scientific publications in the art of refrigeration. The Society is much indebted to former President, Mr. Henry Torrance, for stimulating the issue of a journal, for The Journal is the best means of bringing before the public the work of our Society.

In the course of this year we had the proof of the necessity of making the existence and the work of this Society more generally known. The Fire Department of the City of New York had attempted to issue rules and regulations for refrigerating plants of all kinds, and these proposed regulations had been drawn in such a way that they would have been unnecessarily cumbersome and expensive to owners of refrigerating plants. The Society heard of these intended regulations only a few days before they were to be passed. A committee was at once appointed, and its members appeared before the Commission and succeeded in having many of the proposed rules changed to a more practical form. One aim of our Society must be that no commission appointed by a State or a city to issue regulations for refrigerating plants should do so without asking for the cooperation of our Society, because there is certainly no more competent and representative body of men for such a task in this country.

I believe it is desirable, and even necessary, to establish certain rules for the safe installation and management of refrigerating plants, but they should be practical and reasonable, so as not to hamper and hold back an industry which is of such importance and so indispensable to our country and especially our big cities. We have now a committee on Municipal and State Rules and Regulations for Refrigerating Plants and Refrigerants. I regard it as one of the most important tasks of that committee to work out such regulations for ice plants, cold storage warehouses, breweries, etc., to be used by those States or communities which intend passing regulations. This would secure fair treatment for all parties concerned, and would avoid the possibility of 48 widely different State regulations with "57" more varieties for the different cities.

I wish to add another recommendation. Our Society is entering the twelfth year of its existence, and I believe it to be advisable to commence the compilation of a chronicle or history of the Society. There are still many charter members among us who have taken an active part in the development of the Society, and who would undoubtedly be willing to assist in collecting the facts.

In accordance with the rules we had a meeting in the West this year, which was held in San Francisco in conjunction with the International Engineering Congress. Thirty-five members were present at the meeting—not a large attendance, but satisfactory considering the small membership on the Pacific Coast and the great distance from the domiciles of the great majority of our members. I believe all who attended the meeting considered the effort well worth while. A number of good papers were presented, the discussions were interesting and the social part of the meeting was most enjoyable. The preparations for the meeting were in the hands of a committee of members in San Francisco and Los Angeles, and we are greatly indebted to them, and especially to their chairman, Mr. George H. Geisler, for their work.

On our trip to California we had the pleasure of visiting several raw water ice plants, installed during the last two years. The results obtained in all of these plants are splendid, and the owners are of opinion that in future distilled water ice plants will be installed only where special conditions demand them. I believe all of you have had the same experience here in the East, and the fact that several new raw water can ice systems have been developed, and have proved their reliability in practical service, is of the greatest importance from an engineering and an economical standpoint. I regard it the greatest progress made in a very long time in our branch of engineering.

With the general introduction of the raw water ice plant, a new impetus has been given to the installation of plants of high economy. With distilled water ice plants there was no great incentive for fuel economy, because the amount of steam used by single-cylinder Corliss engines was just sufficient to supply the necessary distilled water. A coal consumption of 1 ton per 6 tons of ice manufactured was regarded as a good performance. In plants of large capacity the coal consumption has been reduced by the installation of compound condensing engines and multiple evaporators, but even in such plants a ratio of 1:10 is a good average.

Raw water ice plants, however, either operating with superheated steam of 180 to 200 pounds of pressure and poppet valve or uniflow engines, or using internal combustion engines to drive the compressor, have a fuel consumption of less than one-half of the above figures, and not only for the large but even for the smaller ice plants. In the future the fuel cost in the manufacture of ice will become a small factor, for average conditions it may be between 20 and 30 cents per ton of ice and for favorable conditions it may go below 10 cents.

This reduction in the fuel expense will decrease the cost of production and an increase in the use of manufactured ice will surely follow which will be beneficial to all concerned. The public will receive cheaper ice, the ice consumption will increase, the owners of ice plants will have a broader and more stable market, and the engineers and manufacturers will do more business.

But the greatest advantage I see to the refrigerating engineer, in the increasing demand for plants of high economy, is the encouragement and necessity for better engineering and better management of ice plants, which will thus enable us to contribute our share to the great movement of the day—Efficiency.

In closing my address, allow me, gentlemen, to use the words of late President Edgar Penney. In his address in 1908 he expressed my own views and feelings so completely and so well that I think it better to repeat them than to give you a poorer substitute. President Penney said:

"Permit me to call attention to the unusual excellence of the papers to be presented at our professional sessions, and to suggest, owing to the length of our programme, that all discussion be well considered and to the point, strictly following the subject matter of the paper.

"I trust and request that all personalities and irrelevant remarks be omitted. Please do not understand me to mean that personal opinion should not find expression—only the manner of presentation.

"This is a meeting of engineers for the purpose of personal interchange of knowledge pertaining to our profession, and should be conducted in harmony and with dignity."

The report of the Committee on Suggesting a Standard Basis for Finding the Fuel Economy of Steam-Driven Can Ice Manufacturing Plants was now presented by N. H. Hiller, its chairman.

The proposed suggestions of the committee were taken up seriatim. With slight modications, all the suggestions of the committee were accepted, except paragraphs 8 and 9. The seeming impossibility of the meeting to agree upon the text of these two paragraphs resulted in a motion prevailing referring the entire regulations to the next annual meeting of the Society, with instructions to the Secretary to send a copy of the proposed rules to each member of the Society, and at the same time request suggestions as to eliminations and elaborations. The revised report is as follows:

REVISED REPORT OF COMMITTEE ON SUGGESTING A STANDARD BASIS FOR FINDING THE FUEL ECONOMY OF STEAM-DRIVEN CAN ICE MANUFACTURING PLANTS

1. *Preliminary Run*—After the contractor has declared that the plant is ready to operate, there shall be a preliminary run of at least 10 days prior to the date on which the test is to be started, for the purpose of standardizing the work of the plant and getting it into satisfactory and normal condition before the starting of the test. During the last 72 hours of his preliminary run the ice shall be pulled regularly and at a rate not less than the capacity specified in contract.

2. *Duration of Test*—The test run shall follow directly after the preliminary run, and its duration shall be at least five days, or 120 consecutive hours.

3. *Condition of Plant*—At the time that the test run is started the conditions of the plant are to be carefully noted, especially the temperature of condensing water, temperature of brine and the condition of the ice in the tank. At the termination of the test, conditions should approach those at the start as closely as possible, particularly in the temperature of the brine and the condition of the ice in the tank.

4. *Readings*—The readings are to be taken every 30 minutes, and are to consist of at least the following:

a. Temperature of atmosphere taken outside of the building in the shade.

b. Temperatures of condensing water at condenser inlet and outlet from machine.

c. Weight of feed water.

d. Temperature of brine in ice tank about midway between top and bottom.

e. Temperature of feed water entering boiler.

f. Pressure, temperature and dryness of the steam from the boilers.

g. High and low pressures of the refrigerating medium.

h. Speed of refrigerating mechanism.

In addition to these readings any other temperatures, pressures and data that may be thought desirable by the parties at interest may be taken, but it is considered essential that the above readings be recorded. Logs to be kept in duplicate and to be signed by the representatives of owner and contractor, each retaining one copy.

5. *Method of Pulling Ice*—The ice is to be drawn at regular intervals

throughout the 24 hours. For example, assuming that 100 blocks are to be drawn in 24 hours, then this would necessitate pulling the ice at the rate of one block every 14.4 minutes, or two blocks every 28.8 minutes.

6. *Over Capacity*—The contractor may overdraw capacity each day as much as he deems his plant capable of handling, providing that the temperature of the brine is not increased in the tank and that the ice drawn is of full weight, and does not contain cups exceeding those described in Article 7. The determination of the capacity shall be based on the average of the ice pulled for the entire test.

7. *Weight of Blocks and Limit of Cups*—The can fillers shall be adjusted so that the blocks pulled during the test shall not weigh less than 2 per cent under or more than 2 per cent over that specified in the contract as the standard weight per cake. The size of cups shall be limited to an opening that will hold not more than one quart of water. For the purpose of ascertaining the weight of the ice drawn, every tenth pull shall be weighed during the test period, and an arithmetical average of same shall be taken and used as the average weight of the ice blocks throughout the test.

8. *Economy*—When it is desired to ascertain the economy of the plant, such determination shall be made during the period of the capacity test.

This economy to be based upon the ratio between the pounds of ice produced and the weight of feed water used; corrected for the heat equivalent of the steam and feed water as compared with the heat in dry saturated steam at 100 pounds gauge pressure from water at 60 degrees F., this being the standard heat value for one pound of steam to be used for the purpose of economy trials.

The feed water shall be weighed throughout the whole period of the test.

9. *Preparation for Test*—If the water temperature varies to any marked degree from that specified in contract, the amount of the allowance for such variation shall be determined, and the parties at interest shall agree upon such allowance prior to the starting of test.

It will be the duty of the purchaser to see that the apparatus is properly cleaned and made ready for test. All condensers, either steam or ammonia, and other cooling coils, shall be properly cleaned and scaled. The boiler and setting is to be put in proper condition and thoroughly cleaned. Any insulation of steam or exhaust or other exposed pipes to be completed by the party furnishing the same under the contract. The purchaser will also furnish the scales or make such other arrangements as are necessary for weighing the boiler feed water and ice. All other apparatus necessary for making test, such as thermometers or pressure gauges not necessary for the ordinary operation of the plant, are to be loaned by the contractor and disconnected by him on completion of test. Thermometers and gauges used throughout this test to be calibrated before and at the completion of test.

10. *Leaks*—Any leaks in steam, exhaust, blow-off or water lines that would have any effect upon the economy or capacity of the plant are to be properly repaired by the party furnishing the same before the test is started.

11. *Arbitration*—In case of controversy it is recommended that a committee of three arbitrators be selected. The contractor and purchaser to choose one arbitrator each, and these arbitrators to decide between them upon an umpire or third arbitrator, and any decision arrived at by this

board of arbitrators shall be final. The expense of this board to be shared equally by the contractor and the purchaser, and all these arbitrators, in order to qualify, shall be members of this Society. In case the two arbitrators fail to agree on a third arbitrator within 30 days of their appointment, then the acting President of this Society shall name such third arbitrator.

12. *Amendments*—The foregoing rules may be amended at any regular or special meeting of the Society, by a vote of the majority of the members present; provided that publication in A. S. R. E. JOURNAL or written notice of such amendment be given the members 30 days in advance of meeting.

Committee on Municipal and State Rules and Regulations for Refrigerating Plants and Refrigerants—The following report was made by its chairman, W. Everett Parsons:

The Committee on Municipal and State Rules and Regulations for Refrigerating Plants and Refrigerants begs to report as follows:

Since the last annual meeting of our Society the committee has held several meetings at the headquarters of the Society and has also attended meetings of the Board of Aldermen of New York City, and has been instrumental in having a number of modifications made in the regulations for New York City as drafted a year ago.

The regulations as finally adopted and passed by the Board of Aldermen of New York City on May 11, 1915, are published in the A. S. R. E. JOURNAL for May, 1915.

Although our committee assisted in bringing these regulations to their final form, the members feel that there is room for improvement, and also that better results might have been accomplished if a committee from our Society had been called upon to assist in the original draft of the regulations. Considering this, it might not be amiss to suggest that the committee be continued with, perhaps, a new membership, who will introduce new ideas, so that an improved set of standard regulations may be formulated in anticipation of the action of other municipalities or States, in order that the interests of the various industries using refrigerating machinery may suffer the least possible inconvenience when such regulations go into effect.

Regulations of this kind do not always originate from the same source, nor is the idea always conceived with the same objects in view. For instance, in New York City the idea originated in the Fire Prevention Bureau, with the chief object the protection of the men from the city fire department while at work where ammonia is used for refrigerating purposes.

Among other things our committee arranged to have the regulations cover refrigerating machinery and apparatus using other refrigerants besides ammonia. In case of a law, recently enacted in the State of Masschusetts, it would seem that the only object sought was the safety from disruption of ammonia compressors by an excess of pressure.

Here, again, only ammonia as a refrigerant was considered, and a special ammonia safety valve has been devised in an endeavor to meet the requirements of the law. This valve will be described in one of the papers to be presented at this meeting.

The members of the committee from Massachusetts will be able to inform you more fully regarding what has been done and what is anticipated in their State.

As mentioned in Chairman Parsons' report, the Massachusetts section of this committee and others were co-operating with the Board of Boiler Rules of Massachusetts in formulating rules to enforce Chapter 467 of the Massachusetts laws of 1914, relative to safety valves as applied to ammonia compressors.

The report of the Massachusetts section of the committee on Municipal and State Rules and Regulations for Refrigerating Plants and Refrigerants was made by F. L. Fairbanks and was in substance as follows:

The Massachusetts Legislature enacted on May 2, 1914, a bill requiring a safety valve on every ammonia compressor, and specifying that the regulations for governing the construction and application of these valves be formulated by the Board of Boiler Rules, and that the enforcement of the regulations be under the supervision of the Boiler Inspection Department. The law is as follows:

Section 1. It shall be unlawful to use an ammonia compressor unless it is equipped with a safety valve.

Section 2. The Board of Boiler Rules shall, within ninety days after the passage of this act, formulate rules for the size, design, location and piping of safety valves on ammonia compressors.

Section 3. The rules so formulated shall have the force of law, and shall be printed and furnished to those requesting them by the Boiler Inspection Department.

Section 4. Any changes in the rules as formulated by the Board of Boiler Rules shall be made in accordance with section twenty-six of chapter four hundred and sixty-five of the acts of the year nineteen hundred and seven, as amended by section two of chapter three hundred and ninety-three of the acts of the year nineteen hundred and nine.

Section 5. The provisions of this act shall be enforced by the Boiler Inspection Department of the district police, and all persons, firms or corporations violating the provisions of this act shall be punished in accordance with section eighty-six of chapter one hundred and two of the Revised Laws, as amended by section three of chapter three hundred and ten of the acts of the year nineteen hundred and five.

Immediately after the passage of the bill the writer was requested by the Board to assist them in the work of preparing a code in compliance with the law. After a conference it was decided to make a practical test of the different valves on the market, with a view to determining their merits and capacities as well as their defects.

The writer offered the facilities of the Sargent's Wharf power house of the Quincy Market Cold Storage & Warehouse Company for such tests, and suggested that Professor Miller, of the Massachusetts Institute of Technology, and Professor Read, of the Worcester Polytechnic Institute, be asked to form, with the writer, a committee of three to go into the matter thoroughly and report its conclusions to the Board.

This was carried into effect immediately, and preparations were made for not only a thorough working test of well-known makes of valves, but for determining the flow of ammonia through orifices and the resistance to flow in pipe lines, etc.

These tests extended over a period of more than six months, and the results were accomplished by pumping the ammonia gas from three 200-ton shell coolers with a 400-ton compressor, and discharging the gas through a

6-inch header to the suction side of a 1,000-ton compressor, by which means it was possible to get any pressure desired, as well as to get a sufficient degree of superheat to insure having a gas temperature so high that its character, while under observation, would be constant and without danger to a near approach to the saturation point.

The entrance to this header was through a calibrated Napier's orifice, and it was provided with valves, test gages, oil wells for test thermometers, outlets and by-passes for attaching the safety valves to be tested.

Six different ammonia safety valves, of as many well-known makes, were tested under the different conditions, and under a wide variation of pressures, most of the tests being witnessed by members of the Board of Boiler Rules and by members of the A. S. R. E. committee.

The conclusions drawn from these tests were as follows:

First—That iron or steel valves with iron or steel seats were not safe, for the reason that the disc is readily corroded to its seat by a small amount of moisture, two of the six valves tested failing to open at 350 to 375 pounds, although they were set at 250 pounds.

These valves upon being dissembled were found to be rusted solidly to their seats and the winged guide in the port; this was undoubtedly due to the small amount of moisture left by the steam which was used by the maker to test the valve and set the spring, notwithstanding the fact that the rust was perfectly dry when the valve was taken apart after failing to open.

Second—That the cast iron valve discs with winged guides in the port would not stand well the excessive chattering and hammering on its seat, to which most of the valves were subject when opening and closing, this being due to the surging of gas pressure which is always present in the discharge line of a compressor, especially at the higher speeds.

One of the valves broke off its winged guide in the port and wedged the disc so that the line had to be shut off; on examination showing that the metal of both seat and guide, which were of cast iron, were of good material and sound, the fracture being caused by the heavy chatter and pound as it opened and returned to its seat.

Third—That the capacity of all the valves were very low except the pop valve.

Fourth—That the pop valve gave practically no warning of when it was about to blow, and for this reason would be dangerous to any one who might be in the path of the discharge when it opened.

It was also shown that the seat of a pop valve could not stand the blow of the valve when closing and remain tight.

Fifth—That none of the valves, notwithstanding their low capacity, could discharge through a pipe line as tapped in their discharge outlet and maintain their setting and character.

It was shown that a 2-inch valve would require a 6-inch pipe connection to carry its discharge gas the first 100 feet with a proportionate increase for each added length, and this without taking into consideration bends or fittings, which would be necessary in most cases.

Discharge pipe lines with test gages were erected so that this feature could be definitely demonstrated, the line being so constructed that readings could be taken for both 50 feet and 100 feet.

The regulations as written were based upon the knowledge gained from these tests, and it was then found that to comply with the regulations an entirely different form of safety valve from any obtainable would have to be designed. The writer was selected to do this work, inasmuch as the valve then in use in our plants required only minor changes to enable it to meet the conditions as specified. The valve* has been thoroughly tested out and has been accepted, and is now available.

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* The Quincy Market Safety Valve for Ammonia Compressors, A. S. R. E. JOURNAL, November, 1915.

The Massachusetts Safety Valve Regulations are as follows:

MASSACHUSETTS SAFETY VALVE REGULATIONS

DEFINITION OF AMMONIA COMPRESSOR

1. An ammonia compressor is any device which is capable of raising the pressure of ammonia gases by mechanical compression, and which is designed, intended and used for such purpose. The term ammonia compressor, as applied to these rules, is intended to include both single and double acting types, with one or more compression cylinders, whether they be of the reciprocating, rotary or centrifugal type, so called, or a combination of either or all, and will be considered to be that portion of the apparatus beginning at the gas intake connection and extending as far as the first stop valve on the discharge pipe.

DEFINITION OF SAFETY VALVE

2. A safety valve under these rules may be any mechancial device which will automatically prevent a pressure in excess of two hundred seventy-five (275) pounds per square inch gage, which operates by allowing the escape to the atmosphere of the ammonia in excess of that pressure, and which has been formally accepted by the Massachusetts Board of Boiler Rules.

COMPRESSOR CAPACITY

3. The capacity of an ammonia compressor shall be taken as the displacement per minute in cubic feet at the maximum speed at which it is operated.

USE OF COMPRESSOR

4. When an ammonia compressor is used for any other purpose than ice making or refrigeration, the conditions shall be referred to the Board of Boiler Rules.

SAFETY VALVE LOCATION

5. (*a*) Every ammonia compressor shall have one or more safety valves located on the discharge side of the compressor and between the displacing element or elements and the first stop valve, and be of such size and adjustment that the pressure cannot exceed two hundred seventy-five (275) pounds per square inch gage.

(*b*) There shall be no stop valves of any description between the safety valve or valves and the displacing element or elements.

MATERIALS AND DESIGNS

6. Safety valves shall be constructed of materials which are not likely to be corroded by either the gases or liquids with which they come in contact, and shall be designed with an ample factor of safety for the pressure, vibration and operation to which they may be subjected.

SIZES

7. Safety valves which discharge to the atmosphere shall be not smaller than one-half inch (½") nor larger than two inches (2") nominal diameter, and shall be of the direct spring-loaded type, but not what is known as a "pop" valve.

CAPACITY

8. If more than one safety valve is used, their combined capacity shall be taken as the discharge capacity required by paragraph 21 of these rules.

INLET CONNECTION

9. (*a*) Every safety valve shall have a full size direct ammonia inlet connection at least equal in area to the nominal inlet of said valve.

(*b*) When two or more safety valves are used on a common inlet connection, this connection shall have a cross-sectional area equal to or greater than the combined area of the nominal inlets of all said valves.

10. Safety valves shall be installed with as short an inlet connection as practicable to avoid serious vibration stresses, and extra heavy pipe and fittings shall be used.

OPENING PRESSURE

11. Safety valves must begin to open at a gage pressure not to exceed two hundred fifty (250) pounds per square inch; and the lift of the valve disc from its seat at two hundred seventy-five (275) pounds per square inch gage pressure shall be at least one-sixth (1/6) of the nominal diameter of the valve.

TESTS

12. Each safety valve shall be tested and set by the maker in accordance with these rules, before shipment from the factory, and stamped MASSACHUSETTS STANDARD AMMONIA 250-275 LBS., together with the maker's name; and when so tested, set and stamped, the bonnet shall be sealed, preferably with a wire and a lead seal bearing the maker's name and date of sealing.

SEAL NOT TO BE BROKEN

13. After leaving the maker, the seal shall not be broken except for necessary testing and adjustment; and when so broken, the Boiler Inspection Department of the District Police shall be notified, so that the valve may be tested and properly sealed again with the seal of said department.

SAFETY VALVE DISCHARGE

14. (*a*) Every ammonia compressor safety valve shall discharge the ammonia to a location, and in such manner, as not to endanger life or become a public nuisance. The discharge pipe shall be carried at least ten (10) feet above the highest opening in the roof of the building in which the ammonia compressor is located, and to an elevation at least ten (10) feet higher than the roofs of adjacent buildings, ***provided*** such adjacent buildings have an opening or openings in the walls, through which ammonia may enter the interior of the said buildings, excepting as hereinafter provided.

(*b*) A plant having not more than twenty-five (25) pounds of ammonia in its working system may have the safety valves discharge into an areaway unrestricted and open at the top, or into an open space between walls, provided such opening or areaway is at least one hundred (100) square feet in area, and that the discharge pipe is located at least six (6) feet from the nearest wall having an opening into the interior; also that the discharge end of said pipe is turned upward.

(*c*) A plant having more than twenty-five (25) pounds of ammonia in its working system may have its ammonia safety valve or valves discharge as herein provided, if such opening or areaway and such discharge pipe outlet are located, from the nearest wall having an opening into the interior, not less than as many times one hundred (100) feet, and by the square root of as many times six (6) feet, respectively, as twenty-five (25) pounds are contained in the actual working charge: it being intended, *in all cases,* to have an open and unrestricted vertical column of air above the level of the ammonia discharge outlet, the horizontal area of which column shall be not less than one hundred (100) square feet multiplied by the number of times twenty-five (25) pounds are contained in the weight of ammonia in the working system, and of a horizontal radius not less than six (6) feet multiplied by the square root of the same factor.

(*d*) The ammonia gases may discharge into a ventilating stack, the inside diameter of which is at least twelve (12) times the diameter of the ammonia discharge outlet; the first possible outlet of said stack, above the discharge end of the ammonia pipe, being not less than ten (10) feet above the roof, or as before provided in this paragraph; and when such stack is used for this purpose, the discharge end of the ammonia pipe shall be turned upward.

(*e*) An ammonia discharge pipe shall have no connection whereby the gases of discharge may be subject to ignition. No connection shall be made to any pipe which is a part of a steam or plumbing system, or to a pipe or stack which is likely to be seriously corroded or obstructed.

DIFFUSER

15. (*a*) When ammonia is discharged into the atmosphere through a pipe other than a ventilating stack, a diffuser shall be used which shall be attached to the upper end of the discharge pipe. This diffuser shall be of such design and construction that there may be no restriction of the flow of gases, the purpose of this diffuser being to intermix the ammonia gas with air, and prevent the discharge of an undiluted column of ammonia gas.

(*b*) The design of this diffuser shall be such that the diameter of the ammonia nozzle shall be approximately the same as the nominal diameter of the safety valve outlet.

(*c*) When the diffuser is used as the outlet for more than one safety valve, the area of the ammonia nozzle shall be approximately that of the combined area of said valves, based upon the nominal diameter of their outlet.

(*d*) The combined area of the air inlets shall be approximately double the area of the ammonia nozzle; and the total discharge area shall equal the combined areas of the air inlets and the ammonia nozzle.

(*e*) The diffuser shall be attached in a vertical position, and be of such design and construction as to prevent obstruction of its openings by rain, snow, ice or otherwise; and it shall be constructed of material which is not likely to be seriously corroded by either the gases or liquids with which it comes in contact. It must also be accepted by the Massachusetts Board of Boiler Rules, be stamped MASSACHUSETTS STANDARD and with the maker's name.

(*f*) The type of diffuser which will be satisfactory to the said Board is one embodying the principles shown in Fig. 1.

ATTENDANCE AND MEANS OF ESCAPE

16. If the ammonia compressor, while in operation, is under the supervision of an operator in attendance at all times, the safety valves may discharge directly into the compressor room, *provided* there shall be no persons in the building containing said ammonia compressor except those directly connected with the operation of said compressor plant, and provided also that there are ample means of exit for persons who may be in the building in which said ammonia compressor is located.

TAPPING

17. Safety valves must have a one-fourth inch (¼") pipe tapping on the discharge side of the valve disc, which tapping shall be closed with a pipe plug, except when the opening is used for the purpose of testing the valve.

Discharge Pipe

CONNECTION

18. The discharge pipe connection to the safety valve shall be made as flexible as practicable, be connected with the least number of fittings, and in such manner as to prevent damage to the safety valve or its connections, by vibration or other stresses likely to be transmitted.

METALS

19. The safety valve discharge connections shall be of metal or metals not likely to be seriously corroded by the gases or liquids with which they come in contact, iron or steel being acceptable for this purpose.

SIZES AND LENGTHS

20. (*a*) The area of the discharge pipe from a safety valve shall be not less than the area of the nominal discharge outlet of said valve in accordance with the table given herein; not less than this size of pipe shall be continued for the first one hundred (100) feet of pipe or fraction thereof, and for

lengths of over one hundred (100) feet, the pipe shall increase one pipe size for each one hundred (100) feet or fraction thereof.

(*b*) When more than one safety valve discharges into a common discharge pipe, the area of said common pipe shall be at least equal to the sum of the areas of the individual discharge pipes of said valves: not less than this area of the common discharge pipe shall be continued to a point one hundred (100) feet in pipe length or fraction thereof from the farthest safety valve, after which point the pipe shall be increased one pipe size for each one hundred (100) feet additional length or fraction thereof.

(*c*) It is intended that there shall be no reduction or restriction of area of the discharge pipe or pipes, as here given, except as provided in paragraph 15, and that no elbow or tee shall be allowed in any discharge pipe line.

(*d*) Where a change of direction is necessary a bend equal in radius to at least five (5) pipe diameters shall be used; and when more than one safety valve is connected to a common discharge pipe, the connecting fitting shall be a "Y," but not a tee.

(*e*) Safety valves must have their discharge outlets tapped or flanged for pipe connections as follows:

Valve (Inches)	Outlet (Inches)
½	1
¾	1¼
1	1½
1¼	2
1½	2½
2	3

(*f*) The type of safety valve which will be satisfactory to the Massachusetts Board of Boiler Rules is one embodying the principles shown in Fig. 2. In this valve the pressure in the casing above the valve disc shall not be allowed to build up a pressure exceeding five (5) pounds gage to the square inch on the back of the valve disc, this space being vented to the atmosphere through a suitable connection, in accordance with the provisions of paragraph fourteen (14).

MINIMUM SIZES

21. Minimum sizes of safety valves to be used on any ammonia compressor, based upon cubic feet displacement per minute, shall be as follows:

Size of Safety Valve, Nominal Diameter (Ins.)	Allowable Displacement (Cubic Feet)
½	120
¾	280
1	510
1¼	830
1½	1,200
2	2,120

BY-PASS VALVE

22. (*a*) An automatic by-pass valve may be used, adjusted to operate at any gage pressure under two hundred fifty (250) pounds per square inch; but this valve shall not be accepted as the safety valve specified in paragraph five (5).

(*b*) This by-pass valve, when used, shall conform in design, material and construction to the specifications given in paragraph six (6).

EMERGENCY INSTRUCTIONS

Ammonia is colorless as a gas, with a very pungent and penetrating odor. Its specific gravity is .589, or about one-half as heavy as air, and therefore it has a tendency to rise. Advantage may be taken of this property, and wherever ammonia is escaping into still air, the better air will usually be found at the lower levels.

This feature should not be depended upon, however, for the reason that ammonia gas mixes with moving air readily and in proportion to the humidity, the moisture in the air absorbing the ammonia gas and carrying it to all points affected by the air movement.

If a helmet is not at hand, a thick towel saturated with vinegar, and fastened over the mouth and nose, will enable the operator to remain in the gas until the acid of the vinegar has been neutralized by the alkali of the ammonia, the time depending upon the amount of vinegar the towel will hold, which is generally from two to five minutes. A gallon of strong vinegar should be kept on hand for this purpose.

Ammonia causes irritation and congestion of the lungs and bronchial tubes, violent coughing, vomiting, etc., and, when breathed in sufficient quantity, suffffocation and death.

In case of partial or total suffocation by ammonia gas, remove the victim from the gaseous atmosphere at once, carrying him quickly to fresh, moving air if possible, and immediately give manual artificial respiration. Do not stop to loosen clothing, or otherwise lose time, as every moment of delay is serious.

Call the nearest doctor immediately.

If approved mechanical devices for artificial respiration are at hand, they should be used; but the manual method should be applied at once, until the mechanical device is ready to apply or the doctor arrives to take charge of the case.

As soon as possible after starting artificial respiration the clothing should be loosened. as well as any other restriction to the breathing or circulation of the victim.

Oxygen should be given as soon as possible, to displace the ammonia gas in the lungs, the use of the oxygen to be discontinued as soon as the patient shall have revived and received plenty of fresh air.

Artificial respiration should be kept up for a long time, if necessary, cases of resuscitation being on record where favorable results were not obtained until after two hours had passed.

If natural breathing stops after being restored use artificial respiration again.

Do not give the patient any liquid by mouth until he is fully conscious, after which he should be given mild stimulants and mildly acid drinks (vinegar and water being excellent), to neutralize the ammonia absorbed by his system.

Give him at all times plenty of fresh air and keep the body warm.

The handling of liquid anhydrous ammonia is attended with very much greater danger than the gas, for the reason that a portion of the liquid instantly becomes strong gas, which is constantly increasing, while the liquid itself, if it touches the flesh, produces a condition, by freezing, which is much the same in effect as severe burns caused by sulphuric acid or scalding water. This is particularly true where the liquid reaches the flesh through the clothing, as the clothing acts like a reservoir much the same as with scalding hot water. For these reasons, a leak of the liquid should never be approached unless the body is protected by clothing enough to prevent the low temperature of the evaporating liquid from reaching the flesh; there should also be over the clothing a liquid-proof covering which will prevent the liquid ammonia from soaking through. Ordinary woolen clothing covered by a rubber coat and rubber boots will answer for the average small leaks.

In addition to this, the face, eyes and lungs must be protected against both liquid and gas, preferably by an approved liquid and gas-proof helmet, which will provide air or oxygen for breathing for a sufficient length of time to enable the operator to get to the trouble, apply the remedy and get back to a place of safety.

It is especially dangerous for a layman to attempt to handle a leak of liquid ammonia.

One of the safest methods is to cover the leak with as large a stream of cold (not hot) water as possible, until the proper valve can be shut and the leak has stopped. This is also true of a gas leak if it can be located and cannot be handled otherwise.

A fire hose with the stream of water covering the leak will absorb the ammonia readily, and is one of the best methods of handling a serious leak when other means fail.

The following four papers were also presented at this session:

The Flow of Superheated Ammonia Gas through Orifices. Edward F. Miller, Boston, Mass.

Quincy Market Safety Valve for Ammonia Compressors. F. L. Fairbanks, Boston, Mass.

A New Style of Steam Safety Valve. G. H. Clark, Boston, Mass.

The Loss of Ammonia at Compressor Stuffing Box. Louis Block, New York, N. Y.

The session then adjourned.

Third Session, Monday Evening, December 6

The third session of the meeting was called to order at about 8 o'clock P. M., and the following papers were presented:

The Testing of Thermal Insulators. H. C. Dickinson, Bureau of Standards, Washington, D. C.

Preliminary Measurements of the Specific Volume of Anhydrous Ammonia. D. R. Harper, Bureau of Standards, Washington, D. C.

The Work of the Bureau of Standards on Constants of Refrigeration. H. C. Dickinson, Bureau of Standards, Washington, D. C.

Dr. S. W. Stratton, Chief of the Bureau of Standards, also made a few remarks on the cooperation prevailing among the various physicists, chemists, etc., employed by the Bureau of Standards, which he felt was largely responsible for the splendid work done by the Bureau.

The session then adjourned.

Fourth Session, Tuesday Morning, December 7

The fourth session of the meeting was called to order at about 9:30 o'clock A. M. and the following papers were presented:

The Refrigerating Engineer. John E. Starr, New York, N. Y.

The Present Status of Brewery Refrigeration. Peter Neff, Canton, Ohio.

Boiler Room Economies. F. E. Matthews, Leonia, N. J.

Apparatus for Tapping Ammonia Lines under Pressure. F. W. Frerichs, St. Louis, Mo.

The session then adjourned for luncheon at the Engineers Club.

Fifth Session, Tuesday Afternoon, December 7

The meeting was called to order at about 2 o'clock. The following papers were presented:

Effect of Velocity and Humidity of Air on Heat Transmissions through Building Materials. J. A. Moyer, State College, Pa.

Steam Economy of an Absorption Refrigerating Machine. Clarence W. Vogt, Louisville, Ky.

Temperatures of Fruits and Vegetables in Transit in Refrigerator Cars. S. J. Dennis and W. E. Mosher, U. S. Department of Agriculture, Washington, D. C.

At the Tenth Annual Meeting in 1914, F. L. Fairbanks, Boston, Mass., presented a paper on "Design, Construction and Operation of a Thousand Ton Ammonia Compression Refrigerating Machine," and, as time would not permit a discussion on it when presented, President Doelling permitted a discussion on it at this meeting.

The session then adjourned.

Sixth Session, Wednesday Morning, December 8

The sixth and final session of the meeting was called to order at about 9:30 o'clock A. M. and the following papers were presented:

Equations for Ammonia Based on New Experimental Material. Frederick G. Keyes, Hoboken, N. J.

A Theory on Cooling Towers Compared with Results in Practice. B. H. Coffey and George A. Horne, New York, N. Y.

Recent Developments in the Study of Corrosion in Concrete Buildings and Pipe Lines. Morgan B. Smith, Detroit, Mich.

Ammonia Condenser Data. Henry Torrance, New York, N. Y.

The report of the Resolutions Committee was the next order of business. It follows:

REPORT OF THE RESOLUTIONS COMMITTEE

The American Society of Refrigerating Engineers and the engineering profession has sustained a distinct and irreparable loss through the deaths of

A. P. Criswell,
J. E. Quigley,
W. J. Webber,
B. F. Daly,
Thomas L. Rankin,

which are recorded with profound sorrow.

It is therefore fitting that the members of the Society, in convention assembled, pause in their deliberations to pay tribute to the memory of the departed.

These men who have contributed much to the advancement of the art of refrigeration and the welfare of the Society and will always be remembered with respect and affection. We feel keenly the loss of their wise counsel and warm companionship and extend our sincere sympathy to the family and business associates of the deceased.

Respectfully submitted,
(Signed) J. F. Nickerson, *Chairman.*
F. E. Matthews.
Karl Vesterdahl.

Before adjourning, a resolution prevailed extending a vote of thanks to Retiring President Doelling for his valued services to the Society.

The meeting then adjourned.

W. H. Ross, *Secretary.*

Members and Guests in Attendance at Eleventh Annual Meeting

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THE BANQUET

The annual dinner of the Society was held at the Hotel Martinique, New York, N. Y., on Tuesday evening, December 7. About one hundred members and guests enjoyed the occasion. After the dinner the following two illustrated lectures were presented:

Safety Pays. Dr. William H. Tolman, Director of The American Museum of Safety.

The Cork Industry. H. W. Prentiss, Jr., Pittsburgh, Pa.

Addresses were also made by the following, with John E. Starr acting as toastmaster: President-elect Vilter; H. C. Dickinson, Washington, D. C.; J. H. Stone, Palamos, Spain, and Ex-President Doelling.

The Annual Dinner, 1915

A. S. R. E. Journal

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PUBLISHED BI-MONTHLY BY

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

154 NASSAU STREET, NEW YORK, N. Y.

Edited by the Secretary, under the direction of the Publication Committee.

Reprints from this publication, which is copyrighted, may be made on condition that the full title of paper, name of author and date of presentation to the Society are given.

Subscription price, United States and dependencies, $3.00 a year; Foreign countries in Postal Union, $4.00 a year.

Entered as second-class matter, February 11, 1915, at the Post Office at New York, N. Y., under the Act of August 24, 1912.

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1916

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A. S. R. E. Journal

VOL. 2 MARCH, 1916 NO. 5

BOILER ROOM ECONOMIES*

BY F. E. MATTHEWS, LEONIA, N. J.

We are living in an age of alleged engineering efficiency. Although the practical application of science to engineering has been the most important factor in the development of modern civilization, of which refrigerating engineering is justly entitled to a lion's share for supplying means of provisioning *en route*, engineering should not clothe itself in too much arrogance until it has been able to attain an absolute thermal efficiency in prime movers of more than the usual 5 to 10 per cent from coal pile to crank shaft, and more than ½ of 1 per cent from coal pile to incandescent light.

The step preceding the prosecution of any undertaking should be not only to determine whether the particular undertaking proposed is worth while, but to what extent it is worth while. That efforts for higher degree of economy are worth while is clearly shown in Fig. 1. While, as the diagram shows, the boiler room is to be held responsible for less than half of the thermal difference between the heat required to produce, and the heat equivalent to a horsepower-hour, the possibilities of increased economy with comparatively small outlay seem greater than in any other part of the plant.

Unfortunately, on account of the number and magnitude of disagreeable features, the average steam-boiler plant is too frequently looked upon as a much-to-be deplored, though possibly necessary, evil; a thing to be endured once unkind Fate has wished it onto the unforunate owner, but to be personally avoided except in cases of the most dire extremity. This feeling is probably most prevalent in the office, which feels that it purchases exemption from personal contact by paying its numerous bills. Nor is this the extent of the feeling of unfriendliness. It is often to be found to no slight degree in the operating force itself, sometimes apparently including the chief engineer.

It has been said that "The main reason for tardiness in boiler improvement probably lies in the fact that educated engineers are reluctant to do the dirty and disagreeable work involved in boiler tests."[1]

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* Read at Eleventh Annual Meeting, New York, N. Y., December 6, 7 and 8, 1915.

[1] The Transmission of Heat Into Steam Boilers, by Henry Kreisinger; Bulletin 18, of the Bureau of Mines, from which the author quotes freely in this paper.

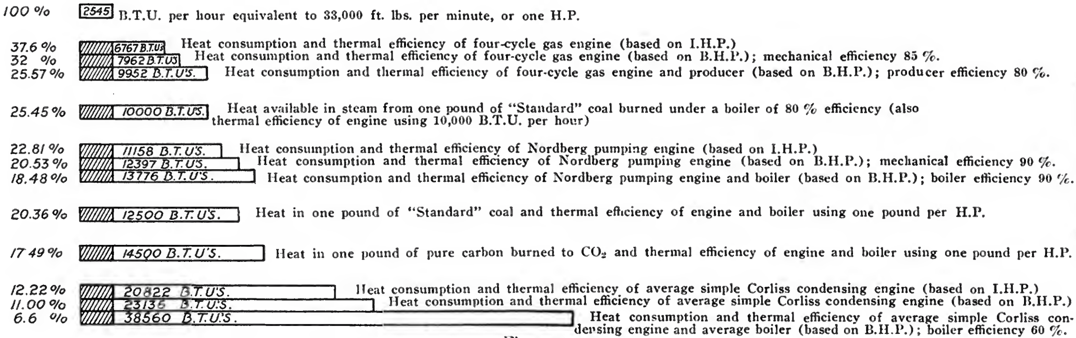
100 %
2545
B.T.U. per hour equivalent to 33,000 ft. lbs. per minute, or one H.P.
37.6 %
6767 B.T.U's
Heat consumption and thermal efficiency of four-cycle gas engine (based on I.H.P.)
32 %
7962 B.T.U's
Heat consumption and thermal efficiency of four-cycle gas engine (based on B.H.P.); mechanical efficiency 85 %.
25.57 %
9952 B.T.U'S.
Heat consumption and thermal efficiency of four-cycle gas engine and producer (based on B.H.P.); producer efficiency 80 %.
25.45 %
10000 B.T.U'S.
Heat available in steam from one pound of "Standard" coal burned under a boiler of 80 % efficiency (also thermal efficiency of engine using 10,000 B.T.U. per hour)
22.81 %
11158 B.T.U'S.
Heat consumption and thermal efficiency of Nordberg pumping engine (based on I.H.P.)
20.53 %
12397 B.T.U'S.
Heat consumption and thermal efficiency of Nordberg pumping engine (based on B.H.P.); mechanical efficiency 90 %.
18.48 %
13776 B.T.U'S.
Heat consumption and thermal efficiency of Nordberg pumping engine and boiler (based on B.H.P.); boiler efficiency 90 %.
20.36 %
12500 B.T.U'S.
Heat in one pound of "Standard" coal and thermal efficiency of engine and boiler using one pound per H.P.
17.49 %
14500 B.T.U'S.
Heat in one pound of pure carbon burned to CO₂ and thermal efficiency of engine and boiler using one pound per H.P.
12.22 %
20822 B.T.U'S.
Heat consumption and thermal efficiency of average simple Corliss condensing engine (based on I.H.P.)
11.00 %
23136 B.T.U'S.
Heat consumption and thermal efficiency of average simple Corliss condensing engine (based on B.H.P.)
6.6 %
38560 B.T.U'S.
Heat consumption and thermal efficiency of average simple Corliss condensing engine and average boiler (based on B.H.P.); boiler efficiency 60 %.

Fig. 1

If this is true, as it probably is, even of more or less scientifically conducted boiler tests that should be inherently attractive to engineers, because of the interesting results they give, a considerable per cent of which are worked out by manufacturers under the spur of close competition, what of the every-day operation, when the chief engineer, usually the only man who is even supposed to know anything about the technical side of such problems, is not only not technically trained, or if so has not at his command or does not employ, apparatus and systems by which he can follow the actions taking place daily and hourly in the boiler room, the economy, as well as the safety, of which is placed entirely in his hands?

To those of a scientific turn of mind, or at least to those to whom too much familiarity has not bred contempt, a boiler room is a laboratory in which a never-ending series of experiments is conducted, the incidental by-product of which is power. If only incomplete, or otherwise inadequate, records of these experiments are kept, or if they are scientifically and systematically kept, but not digested and intelligently interpreted into such practical terms as boiler-horse-power, equivalent evaporation, efficiency, and dollars and cents, the thing of economic interest is lost and only the by-product power, available at an unknown cost, remains.

To attain the best results it is obviously of fundamental importance that the director of the experiments have at least a working knowledge of the natural laws governing the physical and chemical actions involved. While all degrees of success and failure reward the efforts of the man who has not these important assets, that he would attain better results had he possessed them, cannot be challenged. On the other hand, the man who has only this knowledge, but does not know how a shovelful of coal should be spread, and cannot tell by looking at a fire where it should be spread, and it might almost be added cannot pick up a shovel and land it there, may not be in any better position to attain the maximum efficiency than the man who lacks the technical, but possesses the equally necessary practical, knowledge. The appallingly wide margin of loss to be overcome furnishes sufficient motive for its conquest, but the formidability of the undertaking must not be underestimated.

Laboratory research has decided advantages over that conducted in boiler rooms in that it has at its command clean apparatus, means of accurately determining the exact quantities of practically pure materials dealt with and means of maintaining comparatively constant working conditions. In the boiler room heating surfaces are seldom, if ever, clean, both the coal and water contain numerous impurities in varying percentages, and there are always present many variable disturbing outside influences tending to vitiate results, not the least among which may be mentioned the constantly varying temperature, pressure, and humidity of the atmosphere. Even in the adjoining laboratory, the engine room, which usually has the advantage of more intelligent supervision, less constant vigilance is required, for there changes in working conditions are less rapid.

When engine valves, for example, are once properly adjusted engine efficiencies remain practically constant for a considerable time when not prejudiced by the falling off of steam pressures or superheat for which the boiler room is responsible.

As for safety, incidentally, the breaking of an engine part can at most cause no worse a disaster than a fly-wheel explosion, while not only the breaking of a part, but the slightest neglect of the frequently not too responsible water tender, may result in a boiler explosion liberating more energy and producing far more disastrous results. The foregoing digression may be labeled a plea for more intelligent supervision in the boiler room. The common custom of graduating men from the firing room to the engine room may, or may not, tend to relieve the feeling of antipathy for its surroundings and lack the ability to cope with its problems. Be this as it may, it does not affect the financial loss which the owner must meet as the price of inefficiency when it occurs.

Scientific boiler-room management, viewed from a commercial laboratory standpoint, should have for its first object the determination of the exact working conditions under which the investment of a dollar in fuel and operating expense can be made to yield the greatest possible return in heat transported to the engine room in the vehicle steam. The greatest item of boiler-room expense, cost of fuel, will have to be reduced principally through the design of more efficient boilers, and the devising of more scientific methods of operation. This can be done only through greater familiarity with actual specific working conditions and a better understanding of the natural laws in accordance with which the physical and chemical actions and reactions involved take place.

With coal of a given quality at a fixed price, the most important factor entering into the cost of fuel is the completeness with which the two actions, heat production and heat absorption, are carried out. Heat production through that particular exothermic chemical reaction known as combustion, and the absorption of heat so liberated in the furnace gases, by the walls of the metallic containers unscientifically known as boilers, into the water to be evaporated.

Combustion, Table I, is the aggregate of the chemical reactions by which the constituents of the fuel combine with the oxygen of the air with the liberation of heat and light. When the composition of the fuel is known it is a simple matter to determine mathematically or by calorimetry just how much oxygen will be required for its complete oxidation and just how much heat will be evolved. The problems resulting from firing a wet mixture of coal and slate onto a grate which not only allows a considerable portion of it to drop to the ash pit unburned, but in the general case admits either more or less oxygen than is required for its complete combustion, accompanied by its inseparable companions, nitrogen and water vapor; the latter in variable proportions becomes materially more complicated.

The most grievously transgressed mechanical law is that gov-

TABLE I.—DATA ON COMBUSTION

Constituents of Coal	Wt. per Cu. Ft.	Symbol	Atomic Weight	Oxidizes to ——	Composition by Weight	Ratio of Weights	Reciprocal	Theoretically Required Lbs. Air. per		Heat Liberated, B. t. u.
								Lb.	Cu. Ft.	
Carbon	CO 0.07806 CO_2 0.12341	C	12	Carbon monoxide, CO Carbon dioxide, CO_2	C = 12 O = 16 C = 12 O = 32	$\frac{1}{1.333}$ $\frac{1}{2.666}$	.752 .376	5.65 11.3	75. 148.	4.450 14.650
Hydrogen	H 0.00559	H	1	Water, H_2O	H = 2 O = 16	$\frac{1}{8}$	.125	33.9	444.	62.100
Sulfur	SO_2 0.1786	S	32	Sulfur dioxide, SO_2	S = 32 O = 32	$\frac{1}{1}$	1.00	4.23	56.	
Nitrogen	N 0.07837	N	14	Inert.	Composition of the Air wt. 0.8071 lb. per cu. ft. at 62° F.			By wt.		By vol.
Oxygen	O 0.08928	O	16		Oxygen, O Nitrogen, N			.236 .764		.218 .737

erning the proportion of chimneys to the boilers which they serve. The initial installations are not usually so bad, but as the requirements arise, one boiler after another is coupled on to the original chimney until the once possibly efficient plant becomes a veritable gas producer, the flue gases being so rich in carbon-monoxide that it has been facetiously said that they might almost be utilized in the cylinder of a combustion engine. This does not mean that it is not sometimes eminently advisable to add additional boiler capacity to originally well-proportioned chimneys. In fact, it can sometimes be done with considerable increase in economy, provided the engineer knows exactly what he is doing.

The function of a chimney is to hold intact a column of hot and consequently light gases of such a height that the superior weight of the outside air will suffice to cause it to flow into the furnace, overcoming the resistance offered by the grates and burning fuel bed. If the chimney be relieved of the duty of overcoming this resistance, which may be accomplished by forced draught applied under the grates, it may be of sufficient capacity to look after two or three times the original boiler-horse-power.

It is obvious that whatever the source of draught, it must be such as to be able to supply somewhat more air than is required for complete combustion and the minimum allowance for "dilution." Any excess supply can be readily cut off by the dampers as the results of stack gas analysis may dictate.

The effect of insufficient air to produce complete combustion is suggested by the fact that, whereas 14,540 B.t.u. are liberated in the burning of one pound of carbon to carbon-dioxide, only 4,350 B.t.u., or less than 30 per cent is liberated if the oxidation process only reaches the point of producing carbon-monoxide. This loss may be due to lack of sufficient air supply, but it may also arise through conditions under which sufficient air may be admitted to produce complete combustion, but not at the proper time or place. In this event, not only the loss due to the liberation of only 30 per cent of the heat of combustion of the coal has to be reckoned with, but the refrigerating effect of the air not utilized, as well.

A single example will suffice to illustrate. For the combustion of a pound of 14,000 B.t.u. coal 12 pounds of air will be required. Disregarding the water vapors formed by the hydrogen content of the coal, there will be about three pounds of carbon dioxide formed, and there will be about nine pounds of nitrogen present, requiring, respectively, about .63 and 2.16 B.t.u. per degree rise in temperature. Since the coal liberates 14,000 B.t.u., the ultimate temperature resulting from that amount of heat will be 5,000 degrees Fahr.

Assuming that 50 per cent excess air is admitted for dilution, the ultimate temperature will be materially lower because of the necessity of adding six pounds additional air having a specific heat of .237, making it necessary to expend 4.2 B.t.u. per degree rise in temperature, under which conditions the 14,000 B.t.u. of heat available will heat the gases to a temperature of only 3,333 degrees. If

100 per cent dilution is allowed, instead of 50 per cent, the resulting ultimate temperature will be only 2,500 degrees.

The transmission of heat into steam boilers is a matter that has received considerable well-merited attention in the last few years; unquestionably well merited when it is realized that, as was pointed out a few years ago, "James Watt obtained nearly as good an evaporation per ton of coal and per foot of evaporating surface as is obtained to-day."[2] A decade ago it was even claimed that all types of boilers were capable of practically the same degree of efficiency when kept clean and not forced. These statements would be open to question in the present day of high temperatures, high velocities, heavy per cent overloads and often proportionally high efficiencies.

While carbon-monoxide is a result of incompleteness of heat production, excessive air is a most potent cause of incompleteness of heat absorption. Something of the economic importance of reduction of furnace temperatures due to excess air will be shown presently, for the effect is not only to absorb an amount of heat proportional to the range in temperature through which it is heated, but, what is worse, to reduce the ability of the boiler-heating surface to absorb the remaining heat.

Heat transmission takes place in three distinctly different ways—radiation, convection and conduction—each in accordance with laws of its own not applicable to the other two.

A part of the heat evolved in the combustion of coal, in excess of that expended in raising the temperature of the rest of the fuel up to the kindling point, passes to the boiler by radiation. The rest goes to raise the temperature of the gaseous products of combustion, atmospheric nitrogen entering with the oxygen, water vapors and excess air, which impart it by convection to the boiler-heating surfaces over which they pass. The total heat received by the outside surface of the boiler by radiation and convection then passes through the metallic shell and usual coating of scale by conduction. The following equation gives the heat transmitted by radiation from a hot to a cold surface per square foot per minute, when the hot surface is wholly surrounded by the cold surface:

$$\left[\begin{array}{c}\text{Heat}\\ \text{Btu per}\\ \text{sq. ft.}\\ \text{per min.}\\ (H)\end{array}\right] = \frac{2.66}{\begin{array}{c}100{,}000{,}000{,}000\\ (C)\end{array}}\left[\left[\begin{array}{c}\text{The 4th power}\\ \text{of the absolute}\\ \text{F. temp. of the}\\ \text{hotter surface—}\\ (T^4)\end{array}\right] - \left[\begin{array}{c}\text{The 4th power}\\ \text{of the absolute}\\ \text{F. temp. of the}\\ \text{colder surface—}\\ (t^4)\end{array}\right]\right] \qquad (1)$$

"Radiation is that mode of heat propagation by which heat passes through space from one body to another without any material agency; the two bodies do not come into contact directly nor does a third body or other bodies transmit the heat. In case of a furnace and heating plate of a boiler, the heat from the hot fuel bed and furnace walls flows by radiation through the space filled with gases, without heating the gases appreciably, directly into the boiler plate. The quantity of heat transmitted by radiation would not be lessened if the gases did not fill the space; in fact, it would be slightly greater if there were a vacuum between the fuel bed and the plate."

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[2] The Practical Physics of Modern Steam Boilers.

"The quantity of heat which a boiler receives by radiation in a unit of time will be taken to be approximately proportional to the difference between the fourth powers of the absolute temperatures of the hot parts of the furnace and the sooted surface of the boiler plate."

"Convection of heat from one place to another always implies the motion of a fluid receiving or giving up the heat. Thus, in case of the hot gases and the boiler, the heat is transferred from the gas to the heating plate by small particles moving from the body of the gas to the plate and imparting their heat to the latter by their impact upon it. In other words, convection is the constant interchange of position between cooled particles (molecules) of gas next to the heat-absorbing surface and hotter particles within the body of the gas."

"In modern steam boilers, by far the greater percentage of the total heat absorbed by the heating surface is imparted to it by convection. Furthermore, the gases, as they pass along the heating surfaces, are cooled by convection, and therefore the degree of cooling of the gases before they finally leave the boiler is dependent solely on the activity of convection. Consequently, convection largely determines the efficiency of boilers. The rate at which the heat is imparted per unit of surface depends on the number of gas particles coming in contact with it."

The equation for convection, in which A and B are constants, depending on the nature of the gas and units employed, is:

$$\left[\begin{array}{c}\text{Heat}\\ \text{Btu per sq. st.}\\ \text{per minute}\\ (H)\end{array}\right]=A\left[\begin{array}{c}\text{Difference in absolute}\\ \text{temps. F}^{\circ}\text{. between}\\ \text{gas and metal}\\ (T-t_1)\end{array}\right]+B\left[\begin{array}{c}\text{Density of}\\ \text{the gas}\\ (q)\end{array}\right]\left[\begin{array}{c}\text{Velocity of}\\ \text{the gas}\\ (v)\end{array}\right]\left[\begin{array}{c}T-t_1\end{array}\right] \quad (2)$$

Velocity is used in the above equation simply as an index to rate of agitation, the latter, and not necessarily the former, being responsible for increased heat transmission.

"Conduction of heat is the process by which heat flows from a hotter body to a colder one, when the two bodies are in contact, or from a hotter to a colder part of the same body. Conduction implies no visible or mechanical motion of different bodies or different parts of the same body, but implies a direct contact. In a boiler, the heat passes by conduction from the particles of hot gas that touch the soot coating, into the soot and then through the metal and the scale into the particles of water next to the scale." The equation for conduction is:

$$\left[\begin{array}{c}\text{Heat Btu per}\\ \text{sq. ft. per minute}\\ (H)\end{array}\right]=\left[\begin{array}{c}\dfrac{\begin{array}{c}\text{Average conductivity of the sub-}\\ \text{stance between the dry and}\\ \text{wet surfaces.}\\ (C)\end{array}}{\begin{array}{c}\text{Distance between the two}\\ \text{surfaces}\\ (d)\end{array}}\end{array}\right]\left[\begin{array}{c}\text{Difference}\\ \text{in absolute}\\ \text{temp., F}^{\circ}\text{.,}\\ \text{between}\\ \text{dry and wet}\\ \text{surfaces.}\\ (T_1-t\end{array}\right] \quad (5$$

The object in introducing the above equations is simply to show what are the controlling factors in the three means of heat transmission and approximately to what extent they operate. In applying these formulæ to boiler problems, it must be remembered that the heat generated in the furnace reaches the outside of the boiler-heating surfaces by convection and radiation, after which it is transmitted through the sheets, tubes and scale by conduction. The heat thus conveyed to the inside of the boiler passes to the adjoining film of steam and water by conduction, but immediately bubbles of steam are formed, which, being displaced by water, carry the large quantities of heat represented by the latent heat of vaporization away from the heating surfaces by convection. Fig. 2 is a graphical means of representing the various temperature drops in the path traversed by the heat, and is thought to be fairly representative of practice. This figure shows very strikingly that a wide difference in temperature is necessary to induce the heat to pass from the hot fuel bed, furnace walls and luminous flames by radiation from the hot furnace gases by convection to the boiler-heating surface.

Equation I indicates that heat is transmitted by radiation in proportion to the difference between the fourth powers of the absolute temperatures of the object radiated from and that radiated to. This indicates that the quantity of heat that a boiler will absorb by radiation increases rapidly with furnace temperatures. Assuming the temperature of the soot coating of the boiler surfaces to be 540 degrees F., 1,000 degrees F. absolute, the heat which it will absorb from the fuel bed of the ideal temperatures determined above for the condition of 0 per cent, 50 per cent and 100 per cent air dilution, will be as follows:

PERCENT AIR DILUTION	0%	50%	100%
Increase in temperature of gases	5,000	3,333	2,500
Over temperature of air	100	100	100
Degrees absolute—F. zero	460	460	460
T Absolute temperature of gases	5,560	3,893	3,060
t Absolute temperature of boiler	1,000	1,000	1,000
H Btu radiated	923	229	86
% Btu radiated	100	24.8	9.3

The effect of 50 per cent air dilution in the case assumed is to reduce the heat transmitted by radiation to 24.8 per cent of what it was with the exact theoretical quantity. The effect of 100 per cent air dilution is to reduce the radiation to 9.3 per cent.

It will be understood that these "ideal" temperatures cannot be realized in practice on account of the rapidity with which the heat is transmitted away from the fuel bed by radiation. The less heat absorbed by radiation, the more remains to be absorbed by conduction, but, as the latter is proportional to the difference of the first powers of the temperatures, while the former is as the fourth powers, the total absorption will be materially less in the case of the lower temperatures.

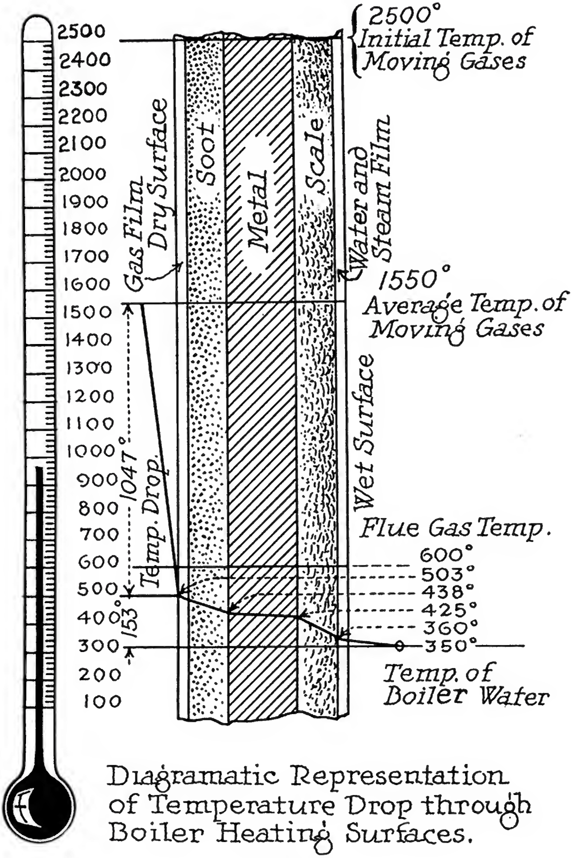
Equation 2 indicates that the rate at which heat will pass by conduction depends, as already stated, on the difference of the first powers of the temperature and also on the density and velocity of the gas, these two factors determining the frequency of impact of gas molecules on the heat-absorbing surface. It will be noted, however, that the density of the gas decreases approximately as its absolute temperature, which offsets to some extent the effect of increased temperature.

Equation 3 indicates that the heat transmitted by conduction is proportional to the conductivity and the difference of the first powers of the absolute tempertures of the dry and wet surfaces and inversely proportional to the thickness of the substance traversed by the heat.

As is graphically indicated in Fig. 2, heat conduction is so rapid through the metal plates of boilers that our principal concern should be not with getting the heat through the plates, or even from the plates into the water, as here heat transfer is greatly facilitated by the presence of the liquid, but with getting heat from the fuel bed and hot gases into the plates. Anything that will tend to bring the flue gases into intimate contact with the heat-absorbing surfaces after allowing sufficient time to elapse for the heat to pass by radiation will act to increase the efficiency of the boiler. Such conditions seem to call for liberal furnace and combustion chamber volumes with a low bridge wall so that the velocity of the burning gases will be slow during the period of radiation; numerous long tubes of small diameter and sufficient draught, either forced or natural, to provide slightly more air than is required under maximum conditions. As the transmission by conduction through the plates and scale is proportional to the difference in temperature and inversely proportional to the resistance offered by these two substances, which increase directly with the thickness, it is obvious that the best efficiency can be obtained when both the tubes and the scale deposits are kept as thin as possible.

Again, returning to the question of air dilution, it is evident that the admission of air which refrigerates the products of combustion reduces heat transmission by both conduction and radiation. This indicates the importance of preventing boiler-setting leaks.

The foregoing has dealt with the production and the absorption of heat. In dealing with the general subject of boiler-room economy, the first place to look for a possible saving is obviously in the coal contract. On what basis and after what investigations were its terms fixed? Were all possible sources of coal supply investigated? Were all the important characteristics of the coal from each of these sources accurately determined, not only by laboratory tests, but also sufficiently long evaporation tests conducted in accordance with accepted rules, by a dependable engineer familiar with the conduction of such tests and alert to the possibilities of the various kinds of



Diagramatic Representation of Temperature Drop through Boiler Heating Surfaces.

Fig. 2

coal? Was each coal fired under conditions most advantageous to its peculiar characteristics? Was that coal purchased which produced the greatest boiler-horse-power per dollar fuel and labor investment? Is it not possible that some other coal making a poorer showing under the test conditions might, with properly adapted boiler setting and draught, produce cheaper steam than the coal purchased? Is the coal so handled that it contains an excess of moisture to be evaporated? Is there an excess of fixed carbon in the ash? Does the fireman handle his fires in the best possible manner? Does he fire frequently and evenly on a not too thick fuel bed? How is it known that this is the case? Are the boiler-heating surfaces clean outside and in? How frequently cleaned? Are the grates adapted to the coal? Are they kept in good condition so that their entire area is active? Does the fireman run water into the ash pits? Is the feed-water as free from scale-forming matter as can be obtained without prohibitive cost? Would fuel loss due to scale pay for water purification? Is this definitely known? Does the feed-water pass to the boiler at the highest temperature that unutilized steam will produce? Are there preventable air leaks, steam leaks or heat leaks anywhere?

In looking for leaks in a manufacturing business one of the first places to be inspected by the old, horse-sense industrial engineer, before the advent of his efficiency successor, was the scrap-heap. The cinder pile, in so far as it goes, is to the power plant what the scrap-heap is to the manufacturing plant, and as such is entitled to its share in the distinction of being inspected. The thermal scrap-pile of a power plant, however, includes not only the cinder pile, which should always be held under suspicion of containing good raw material in the form of fixed carbon, but also the chimney and boiler setting, which are always prodigal in their expenditure of liquid capital, heat, for the sole production of which the steam factory is operated.

Every manufacturer should realize not only that his secondary product, whatever it be, is absolutely dependent on his primary product—heat—and that, like other liquid capital, it should be judiciously and economically expended, but also that he has the responsibility of operating his own *smelter* and his own *mint;* a smelter in which the raw ore—coal—is made to give up more or less of its golden treasure—heat—which in the adjoining mint is converted by steam engines into *power currency* of convenient denominations. The moral is almost too obvious to be stated: Don't watch the expenditure of liquid capital so closely that you forget to analyze the slag.

DISCUSSION

C. W. Vogt—What is the thermal efficiency figured from the heat under the boiler to the heat equivalent to the power produced; did you say 6 per cent?

Mr. Matthews—6.6 per cent.

C. W. Vogt—That was the thermal efficiency of the engine?

Mr. Matthews—No, the thermal efficiency of the engine, based on the indicated horsepower, was 12.22 per cent in this particular case. On the boiler and engine together it is 6.6 per cent.

President Doelling—What steam consumption is assumed per indicated horsepower?

Mr. Matthews—The steam consumption equivalent to 20,822 B.t.u. would vary somewhat with the pressure and feed water temperature. Assuming, for the sake of argument, that the steam pressure is 100 pounds, gauge, and that the engine is charged with all heat above 100 degrees, the steam consumption in pounds per indicated horsepower hour would be 20,822 divided by the heat in a pound of steam above 100 degrees, 1,117 B.t.u., or 18.6 pounds.

F. L. Fairbanks—I would like to say that in our Sargeant's Wharf plant the records show that in the last two years' operation we are obtaining a boiler room efficiency averaging 75 per cent.

C. W. Vogt—That is for the boiler room only?

F. L. Fairbanks—Boiler room only; we have tests of 10½ pounds on the engine.

G. A. Horne—That is with economizers?

F. L. Fairbanks—With economizers, forced draft automatic regulating devices and recorders. As a check, we weigh our ashes out on the same scales each day on which we weigh our coal in. We sample the ash by analysis to show the number of B.t.u. going out in the ash, as well as in the coal coming in. The monthly report to the treasurer shows the cost of evaporating a pound of water. We show the cost of coal in two different ways—for evaporating a pound of water, as well as for the total cost of evaporation, that is, with the labor charge and all other boiler room expenses added. The treasurer has an opportunity to examine the cost both ways—first the coal cost and then the total cost. As the steam is delivered to the engine, the engine room is charged with the steam delivered to it by the boiler plant, so that the two operations are kept entirely separate. The engine room is credited with the primary and secondary exhaust heaters, but not with the economizers. The stack gases run from 175 to 215 or 220 degrees.

President Doelling—I think that 75 per cent efficiency is a remarkable performance.

G. A. Horne—Boiler efficiency varies considerably with the kind of fuel. As Mr. Matthews has pointed out, everything has to be taken into consideration. It would hardly be possible, I think, to obtain 75 per cent with small anthracite coal in small plants. As for the questions Mr. Matthews asks, it is necessary, in connection with small plants, to go considerably into detail in getting the cost of a thousand pounds of steam and weigh carefully all the factors that enter into the case. Now in this territory, particularly on Manhattan Island, real estate is rather valuable and most of the large power plants find it desirable to operate with heavy overloads. As to relative economy of a small plant that shows a higher efficiency with the bituminous coal, possibly with stoker equipment, one is apt to be misled in deciding what might be best for other conditions. To illustrate, I might say that some years ago I made some 40 or 50 boiler tests with the particular point in mind as to which fuel would give the largest amount of steam for the money, all things being considered, but not taking into consideration the fact of overload or the investment in land, as this particular plant was out on the meadows, what we call No. 3 buckwheat of very good grade, carrying between 11,500 and 11,800 B.t.u. on the basis of the dry coal, could be obtained for $1.50 per gross ton. At the same time high grade bituminous coal of about 14,800 B.t.u. value cost $3.00 per ton. It is easy to see that there is a very great difference in the number of B.t.u. per dollar. On careful tests, while you could not get anywhere near 75 per cent efficiency with the anthracite coal, the $1.50 coal, we could make a thousand pounds of steam for about 8½ cents, and with the $3.00 coal it costs about 12 cents. I simply cite this to show that, particularly for plants where they may not require heavy overloads and where the value of the property is not such that they have to crowd their boilers into small space, the investigation of these small sizes of anthracite coal, which always have to be used with mechanical draft, may prove interesting and valuable for small plants. Now those prices do not obtain to-day, unfortunately for some of us who use anthracite coal. That same coal to-day costs nearer $2.00, but it still shows, however, something in its favor on the basis of the fuel cost. I think, as Mr. Matthews has indicated in the series of questions, that a number of such careful considerations is very important for the smaller plant.

Except with very large grate surfaces it is absolutely impossible, from my experience, to get a large overload with the small sizes of anthrcite, although some of the manufacturers of blowers and people who handle apparatus to use in conjunction with that coal will tell you that you can get 50 per cent to 100 per cent over the rating of the boiler. I never could get it; I don't know whether any of you can or not. I have found, in our experience, that we have a good deal more pressure in the ash-pit than the manufacturers of blowing apparatus seem to have in their tables; that may be due to the fact that this small anthracite coal varies greatly in size and we do not always get the size of particles that we are supposed to get. With No. 3 buckwheat, to get the rating of the boiler, it is necessary to have from three to three and one-half inches pressure in the ash-pit, and if you want to realize 50 per cent over rating, which is very difficult to obtain, unless the fuel is of very high grade, you have to have four or four and one-half inches of pressure in the ash-pit.

F. L. Fairbanks—I think the basing of judgment upon the cost of evaporating a thousand pounds, or one pound of water if you will, is very misleading, for the reason that the coal cost is not the total cost. We found that in our plant, using No. 3 buckwheat, if we could control our delivery from certain mines, at $1.80 alongside the dock, we could make an excellent showing, but in figuring up the cost of handling and discharging that coal, carting away ashes, and the extra labor necessary to keep the fires clean—once every four hours, on a forced draft—we found that New River coal at a very much higher price, which showed a higher cost of evaporating per thousand pounds of water, was very much more economical to use, and we discontinued buckwheat entirely. Besides the fuel cost, we now turn in the total cost of evaporating a thousand pounds of water, so that there can be a comparison of the two. These two costs show you just what you are doing and of what practical ultimate value any particular grade of coal is to the plant.

I think the tendency is to expect too much of the smaller plant. In this an efficiency is expected equal to what might be secured only during a test, for which everything is especially prepared. Many do not stop to consider that the ordinary man running a plant 8, 10, 12 or 24 hours a day is subject to practical conditions over which he has no control whatever. We find in our own works that our Richmond street plant with somewhat higher cost is, in a number of ways, a much cheaper plant than our Sergeant's Wharf plant, because the equipment we have in the Sergeant's Wharf plant requires the operation of that plant at a high percentage of its capacity in order to get good results. In other words, during the three winter months we find that our Richmond street plant, which has a very much less expensive equipment and apparently is very much less efficient, is actually a cheaper plant to run than our Sergeant's Wharf plant. In order to maintain the load we transmit some six to eight hundred horsepower, electrically, from our Sergeant's Wharf plant to our Richmond street plant to improve our load factor. It is very easy to criticise the small plants too severely. The continued efficiency of a plant is due as much to general design and equipment as to the operator, who, in a great many cases, has absolutely no control over conditions. He has not the authority to change his equipment. He has got to maintain his output on the conditions given him.

N. H. Hiller—A good many years ago we used to hear of the wooden nutmeg. The anthracite coal man has got an improvement on that; he sells you a lot of slate in No. 2 and No. 3 buckwheat. About five or eight years ago we made tests of No. 2 and No. 3 buckwheat and got an evaporation of about 8½ pounds from and at 212 degrees, to-day we are getting only 5½, simply because we are getting 50 to 60 per cent carbon and the balance ash. Now, we can burn it all right and get good results, and we can force our boilers, but we have got to have the grate surface to do it. We are running about 72 square feet of grate surface on 1,700 square feet of heating surface on a No. 2 boiler, so that our combustion does not run, I think, much over 12 pounds per square foot of grate surface or even lower than that, 10 or 8 will give better results. If you want to get capacity out of a boiler, it is necessary to have grate surface in order to burn the grade of anthracite that is coming through to-day.

The arrangement in our boiler setting is somewhat different from the ordinary setting, as it is designed for the purpose of burning anthracite. It has what is known as a Webster setting, which has an arch over the rear part of the grate. With that there is no difficulty in burning very poor grades of anthracite, in fact what we, in the Anthracite valley, term "colly," which runs as high as 50 per cent and 60 per cent ash and stone.

C. A. Read—In regard to the efficiency of the small plant, I think you will find that a man who has two boilers, and his load runs fairly heavy during the day and not much at night, if he gets 65 per cent efficiency he is doing pretty well. I have made a number of tests in that line where there was a fairly heavy load during the day, when a man has to keep his two boilers going, and during the evening there may be enough to keep one boiler and part of another one. He banks up during the night and you will find he realizes an efficiency of about 50 per cent to 65 per cent. I have seen this obtained while running a mixture of bituminous and No. 2 and No. 3 buckwheat. I notice you gentlemen have experience in burning buckwheat, and I would like to have a meeting with you for the purpose of telling experiences. In regard to the quality of coal at the present time, if any of you are buying what is known as No. 2 nut or pea coal for your house use, I think you will do well to specify that you get it from the screenings from the yard—pea that is broken in shipping. It is this coal that breaks, and not the slate.

F. L. Fairbanks—I don't know whether my friend, Mr. Read, meant to criticise my 75 per cent or not, but perhaps it would be well to say that at our Sergeant's wharf plant we are operating 24 hours a day under both forced and induced draft, and are running 200 per cent in the boiler room all the time. We are averaging 200 per cent to 250 per cent 24 hours a day. We have no banking of fires or starting of fires. The boiler goes in for 30 days under both forced and induced draft. At the present time we are carrying, or were carrying, 7,000 horsepower, with a possible 10,000 horsepower, and we do not have any trouble in doing it. It is not a question of the human element at all, it is absolutely a question of machinery. Operation is automatic from the time the ship comes alongside the dock until the coal goes out over the scales. It is not a question of the human element, it is a question of the amount of money you put into the plant to make it automatic.

G. A. Horne—Mr. Hiller comes from a coal region and ought to know about coal. I hold no brief for the coal men, but I have to say, in fairness, that we have been a little more fortunate in the quality of our anthracite coal up to the present day. It is apt to change, I suppose, but I will say, in fairness to perhaps one or two companies that do give fresh mined coal, even in these smaller sizes, that our coal on analysis of a monthly composite sample—we only analyze it once a month in each plant—averaged last year something like 14 per cent ash on the basis of dry coal, which probably is a good deal better than most of the anthracite coal, because I have personally analyzed coal in the section where Mr. Hiller comes from, and on one occasion it was 41 per cent ash. Our coal averages 14 per cent ash and has a heat value of 12,000 B.t.u. In one of our largest plants, not on test, but under running conditions, we evaporated over 8½ pounds with the Lehigh & Wilkes-Barre Coal Company's coal.

F. L. Fairbanks—It seems to me there is another proposition Mr. Hiller does not count on at all. Being in the coal mining district, he does not have transportation charges to take care of. It makes a whole lot of difference whether you are paying $1.25 or $1.50 freight per ton on the slate that is supposed to be coal, and 25 cents a ton to a stevedore to discharge 40 per cent of non-combustible. We find the prohibitive point is paying for transportation on something you cannot use and also paying transportation on the excess of ash. Our experience has shown that under average conditions we are doing well if the ash going out over the scale is not double what the ash analysis would indicate. Our contract calls for coal absolutely from one mine. The ship is loaded for our plant with that understanding, and under those conditions we can maintain a definite line of operation, but even then, as Mr. Horne says, we have had buckwheat with 20 per cent surface water and buckwheat that has been tested as high as 30 per cent ash. You can readily see that such coal might be very expensive, even if somebody would give it to you.

J. E. Starr—As to the question of burning low grade coal, I had a very striking example the other day. I was in a plant where an efficiency company was trying to burn some No. 3 buckwheat. The plant with No. 1 buckwheat was carrying the load very comfortably under natural draft. They started in to demonstrate that they could burn No. 3 buckwheat under forced draft. The blowers were steam driven. They got along very well for the first half hour, then it came to cleaning the fire. Well, part of the plant had to be shut down, for the simple reason that with the dropping of the steam pressure they could not get the required pressure with their steam-driven blowers, and the thing once started continually got worse. They started with 105 pounds and got down to 60 pounds, and there they stayed. This is simply to illustrate one of the problems. Possibly if these blowers had been driven by some outside source of power, such as electricity, they might possibly have held up.

For most of us, who are consulting engineers, a large share of our business is with small plants. The large plants, such as Mr. Fairbanks and Mr. Horne speak of, have their own force of men who are continually watching these conditions which Mr. Matthews has spoken of, but in the small plant—a 50-ton ice plant, or even a 100-ton ice plant—the average fireman and the average engineer does not know a B.t.u. from a hole in the ground, and I find that the only practical way to get at these people, and to really do something practical in the way of reducing their coal bill, is to induce them to keep books with their boilers just as they keep books with their customers; that is to say, to induce them to put in water-measuring and coal-weighing devices and pit one fireman against another. If you can pay a fireman 25 cents more a day than he is getting, and if thereby he can save you 50 cents a day, you are just

25 cents ahead, or other amounts in that proportion. I have found that in small plants the best and quickest way is to bring it to the owner's attention. He gets his coal bill at the end of the month and says "Oh my, that's a big coal bill, its awful." He cannot account for it; he simply knows in a general way that it is a big coal bill. I have had several cases where, by the expenditure of $100 or $150, I have been able, without going much into detail, to cut off anywhere from 20 per cent, and in some cases 40 per cent, of the coal bill by simply bringing in to the office every day the amount of water pumped into the boiler and the amount of coal used, and then have that posted up in the fire room. Gradually you have two elements; you rouse the fireman's and the engineer's pride in the matter; secondly, by pitting one man against another and paying him a little bit extra for good service and careful attention, you will find that, almost insensibly, he begins to discover why he is not getting good results, as well as the reason why he is getting good results. In other words, he gradually and unconsciously works towards the consideration which Mr. Matthews has put before us.

Here in this Society, the discussion is largely between men who are operating large plants, but I think that the most practical way to get results in a small plant is to induce the owner to put in water-measuring and coal-measuring apparatus and to make the records of their performance a part of the records of the concern, and their daily examination by the manager a duty no less important than the examination of the accounts with his customers.

F. E. Matthews—I wish to ask Mr. Starr, through the Chair, what meter he recommends for metering the water going to the boiler?

J. E. Starr—Any of the well-known metering devices, even a thermal water meter for a little while will hold up, but there are devices on the market which will hold up indefinitely if they are kept in good repair.

F. E. Matthews—How do you find the Venturi meter for accuracy?

J. E. Starr—I find it good. There is also an electrical apparatus, brought out recently, which seems to be doing very well. There is also the tilting meter, which gives very good results. The regular type of water meter is only good for a little while; my experience has been that when it is first put in it works all right, but after a while it evidently indicates that you are pumping into your boiler a good deal more water than you really are.

F. E. Matthews—I would also like to ask Mr. Starr about the plant in the hands of the efficiency company; whether that had mechanical stokers or was hand fired.

J. E. Starr—It was hand fired. The expert understood combustion very thoroughly, and he had under him an expert fireman.

F. E. Matthews—I think it is no uncommon occurrence that the people who are trying to introduce low grade coal find trouble with firemen who do not want the coal to make good. The men in the plant are often directly responsible for the falling down of the low grade coal.

N. H. Hiller—If a certain quantity of high grade No. 1 buckwheat or pea coal is used to start up the fire, you can usually get up your steam pressure all right with that and then maintain it with low grade anthracite. I do not know whether in the test Mr. Starr refers to anything of that sort was tried, or whether No. 3 was to be used altogether.

J. E. Starr—They were trying to use No. 3 altogether. They got through and threw up their hands in about six hours. There remained the question of the quality of the coal, there being no analysis at hand, and whether the failure was entirely due to the fact that the blowers were steam driven, because when the speed was reduced the pressure stayed down. This question has not yet been answered.

C. A. Read—I have not come from Boston and am not trying to speak as often as my friend Fairbanks does, but I think a man should be extremely careful in recommending the use of buckwheat coal, because he runs up against terrible prejudice in the boiler room. I have been up against it several times, and in one case I had to do the firing myself to prove that it could be done. In one case I know of, where it was a failure, I heard afterwards that the engineer had made the remark beforehand that it was going to take him but a little while to show that it was no good, anyway, that it couldn't be done.

In regard to the question of the over all efficiency of the plant, I think a few years ago Mr. Stott made the statement that of the heat of his coal only 2½ per cent appeared as energy on the bus bars of his station, that is the over all from the coal pile to the bus bars of a station. I would like to ask Mr. Starr in regard to the Venturi meter. Suppose a plant is running part of the time 50 per cent over load, and then for 10 or 12 hours during the night it drops down to only 50 per cent load; what percentage of accuracy would he expect to find in the Venturi meter in a case of that kind?

J. E. Starr—I do not know. All plants that I am interested in have practically a steady load all day. They are ice making or refrigerating plants. I have never had any experience with a Venturi meter on a fluctuating load.

C. A. Read—That is where I have had trouble, and with the steam flow meter as well. I have a case now where, in order to have a meter that gives the range during the time the load is on, indicates that they can heat their buildings all night without charge for it at all.

F. L. Fairbanks—As my friend, Mr. Read, knows we have just been through quite an experience with the Venturi meter. We find it limited to certain velocities; in other words, under seven feet per second through the throat and thirty-five feet per second, or thereabouts, the Venturi meter becomes somewhat inaccurate, and this past year we have tried most of the important measuring devices and have found with these that we were evaporating 18 or 20 pounds of water per pound of coal. The treasurer would not accept that, so we had to do something else, therefore last spring we had an entire equipment built for us by the Builders Iron Foundry of Providence, who manufacture the Venturi meter. A meter with their latest indicating, integrating and recording device was installed under their supervision. We then found out several things, first among which was that with a reciprocating pump and an ordinary regulator, the Venturi meter might run 20 per cent ahead or 40 per cent behind, for the reason that the pulsating flow of the reciprocating pump as governed by the regulator, nullified to a great extent the principles of the Venturi meter. We then purchased one of the highest grade centrifugal pumps, driven by a high grade turbine, and had a regulator designed for the turbine which would vary in ounces of differential boiler pressure, so as to keep the curve as flat as possible, and installed a meter with a four-inch stroke. We operated two months, and as our load dropped off we appeared to be evaporating 18 to 20 pounds of water per pound of coal, which could not be correct. Now we have two meter tubes in parallel, so that as our load changes with the season or the weather, we expect to get more accurate results. Should our load drop below the seven feet per second for the tube with the larger throat, we can immediately cut that out and use the tube with the smaller throat, so as to maintain the required velocity through the throat orifice. This, in conjunction with means for keeping the flow of feed water comparatively constant, should enable us to obtain the desired accuracy in metering.

RECENT DEVELOPMENTS IN THE STUDY OF CORROSION OF CONCRETE BUILDINGS AND PIPE LINES*

By Morgan B. Smith, Detroit, Mich.

It has been well said that this is an era of steel and concrete, for truly these two materials are great factors in our present wonderful industrial development. It is equally true that distant transportation of heat and of cold beneath our streets has reached a remarkable magnitude in recent years.

To refrigerating engineers there are presented two phases in the matter of corrosion, namely, the conservation of reinforced concrete structures erected or to be built and, secondly, the conservation of underground piping and other structures. These two phases I intend to discuss, giving such facts as seem to be well established at this time.

We must, however, remember that this study is bringing up new phases almost daily, and that therefore facts of to-day may be discarded to-morrow as false. This is always the rocky road towards progressive thought in which we pass through a sort of inception period generally characterized by numerous theories, few of which survive, followed by a period of comparative surety as facts are brought to light.

Not long ago many of our best investigators felt great concern over the safety of our concrete structures and of underground structures, yet to-day, as the truth is known, we find that much of this feeling of insecurity was needless, or at least comparatively so. I do not mean by this that there is now no danger attendant, but I do mean, on the contrary, that we now know enough of the matter to be able to assure ourselves of the safety of our structures when due precautionary measures are taken.

I may state here that, as time has put it to test, the electrolytic theory of corrosion has gained a firmer hold on fact. Electrolytic corrosion may be instituted in two ways, first by the passage of electric current through the system from an outside source, and, secondly, that set up by so-called auto- or self-electrolysis caused by differences in solution pressures of two dissimilar metals in the system, or by unlike chemical condition of portions of the same metal. As a rule, it may be said that the effect of self-electrolysis is rather insignificant when current electrolysis is present, but is always augmented by current flow from external sources.

We nearly always find active self-corrosion where structures are exposed to a flow of current. It is equally true that at times when external current is absent self-electrolysis is very active and rapid destruction of metal has occurred.

What about our concrete structures?

Probably no better work has been done in the study of the corrosion of concrete structures than that carried out by the Bureau of Standards concerning external current flow, this work not taking into account self-electrolysis. I shall, therefore, briefly outline the main facts brought to light in this study of their subject.

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* Read at Eleventh Annual Meeting, New York, N. Y., December 6, 7 and 8, 1915.

We find three phases to be considered, namely: Corrosion of concrete reinforced structures by flow of current into the reinforcing metal, and thence to the concrete mass, and, secondly, the reverse condition of current flow, and, thirdly, the corrosion of non-reinforced concrete. In the first case we find the metal positive towards the concrete; in the second case it is negative, while in the third we have no metal present, in which case the concrete must act as positive or negative towards some adjacent metallic structure.

Let us consider the first phase, Fig. 1, in which the metal is the anode of the system. When current enters the reinforcing metal it may follow along it or, if the concrete be damp, it may jump from the metal to the concrete, from whence it travels to metallic parts of the building, or to apparatus within the structure. In such a case the metal passes into solution at a rate closely following Faraday's law in most cases, but in others at variance with the law. The concrete here is acting the part of the cathode or negative electrode.

The first effect is the accumulation of the oxides of iron upon the surface of the reinforcing metal, which occupy roughly twice the volume of the equivalent metal. This may give rise to an outward pressure of fully 5,000 pounds per square inch, which will in time

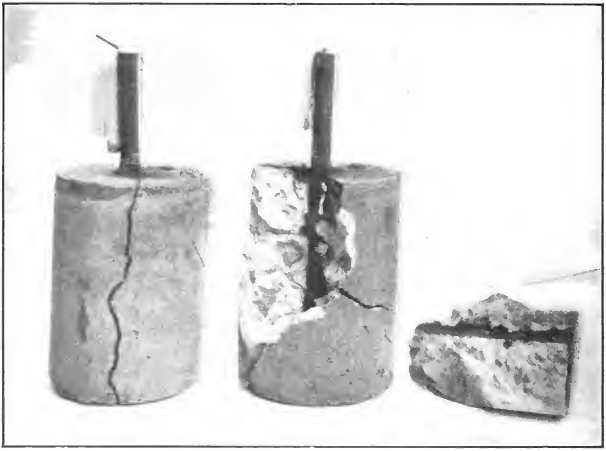


Fig. 1.—High Voltage Anode Specimens—From *B. of S. Bulletin No.* 18
Metal (+), Anode: Concrete (—), Cathode

disrupt the concrete mass. First, fine cracks appear, followed by wider checks and cracks, ending in final fracture. We find, however, that to produce this result consistently there must be present a potential drop through the system of at least 60 volts, and that the temperature must be at least 100 degrees F. The temperature is manifestly a function of the current flow, which in turn depends on the potential drop in the system, and, obviously this depends on the current-carrying capacity of the concrete mass. It is, therefore, evident that materials which will lower the electrical resistance of the concrete should be avoided in the original concrete mix, *especially chlorides.* As concrete ages the moisture content should decrease and the electrical resistance ought to become greater.

There is no doubt that sodium and potassium (alkalies) in the concrete are a detriment to the life of the mass, since, when present, they tend to follow the electric current, migrating toward the cathode. This is clearly shown in Fig. 2, where the concrete is the anode, the metal being the cathode of the system. The alkalies have concentrated around the cathode, rotting the concrete and destroying the bond between it and the metal. Removal of alkalies by electrolysis before the concrete has set entirely removes this dangerous condition, since there is then no alkali content to migrate with the current. This is shown in Fig. 3.

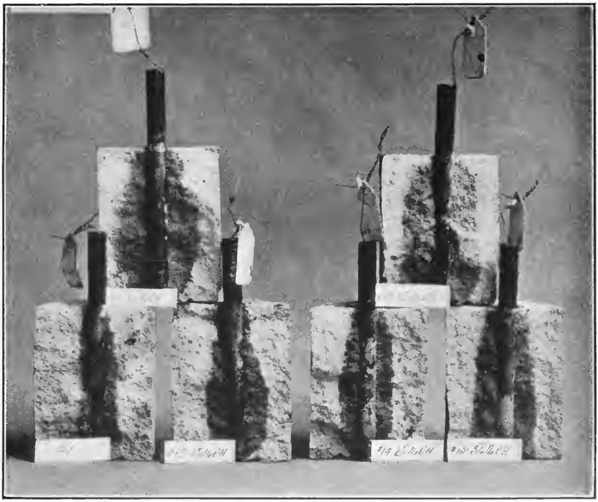


Fig. 2.—Cathode Specimens, Showing Increase of Disintegration Due to Addition of Sodium—From *B. of S. Bulletin No.* 18

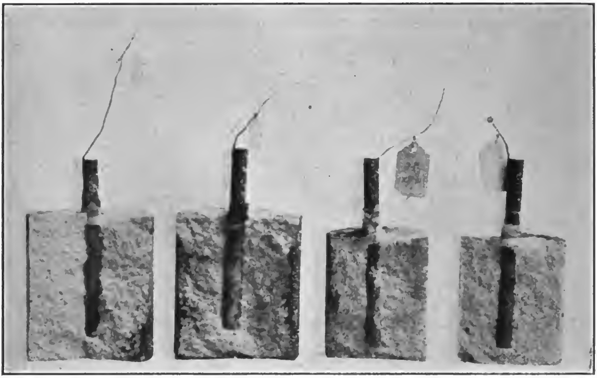


Fig. 3.—Cathode Specimens, Showing Absence of Disintegration Where Alkalies Had Been Removed by Electrolysis—From *B. of S. Bulletin No.* 18

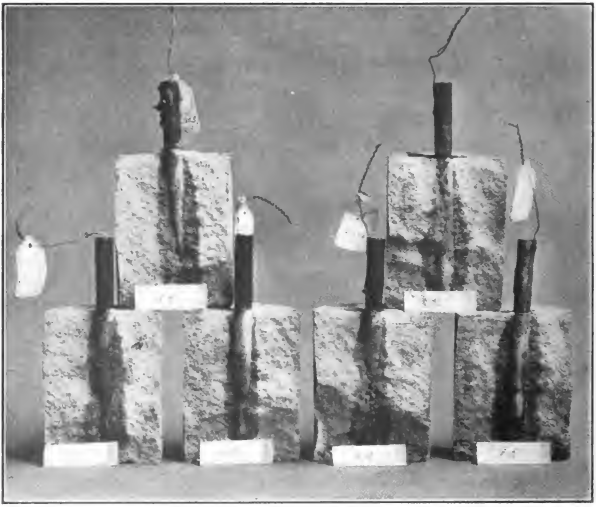


Fig. 4.—Cathode Specimens, Showing Disintegrated Mortar About Electrode—From *B. of S. Bulletin No.* 18. Concrete (+), Anode; Metal (—), Cathode

Let us now consider the case when the concrete is the anode of the system, in which case external current enters the concrete, passes to the reinforcing metal, and thence along it to other metallic parts of the structure or to apparatus situated therein, Fig. 4. Here we find the metal practically unaffected. The concrete, however, shows marked effects, the results being noted in a softening of the concrete mass around the metal, due to concentration of alkalies at that point. The mass, upon drying, becomes hard but brittle, and the bond between metal and concrete is destroyed. Unlike the condition when metal is positive, we find here that the effect on the concrete is manifested at both high and low voltages, being in general in direct proportion to the potential drop through the system.

Before taking into consideration the means available for minimizing these effects we must consider the electrolysis of non-reinforced concrete. Here the effect of current flow is similar to a slow seepage, by which the water-soluble content migrates towards negative portions of the mass or, more generally, towards metallic structures in contact with the concrete. At points of contact between concrete and metal we find the same results as noted above, where concrete is the cathode or negative electrode in the system, namely, a softening of the mass, which becomes hard and brittle on drying. Obviously, grounding of metallic structures in such a case is similar to installing electrodes in the system, and will do harm rather than good.

So much, then, with regard to possible conditions. How shall we proceed to minimize the effects noted in our concrete structures?

First—We should take every precaution to keep potential gradients in the structure as low as possible.

Second—We should coat all surfaces of concrete below ground with an insulating film of some sort, preferably a flexible film which will remain water-tight at all seasons.

Third—We should paint all reinforcing metal in so far as is practicable with an insulating paint of such a nature that its application does not lessen the bond between the concrete and the metal. The practicability of this procedure at present seems somewhat doubtful, although paint manufacturers assure us that they will eventually develop such a material. We have at present at least one such paint, which approaches closely the desired characteristics.

Fourth—We should take great care to keep all direct-current circuits in the building free from grounds.

Fifth—We should provide all pipes, entering or passing through the building, with insulating joints both at points of entrance and exit. If the potential drop around such an isolated portion of piping exceeds 8 volts, a copper shunt, preferably of insulated cable, should be carried around the isolated portion.

Sixth—We should make complete inter-connection, electrically, between all metallic structures within the building, but not until after insulating joints have been provided for all piping entering or passing through the building. Such inter-connected structures must not be grounded outside of the insulating joints.

Seventh—As a precautionary measure it is well to use granite blocks between the footings of our buildings and the soil, since such blocks are, roughly, about 100 times as resistant to current flow as is concrete, especially damp concrete. There will then be far less tendency for the building to pick up stray currents. The use of granite is especially recommended when marked potential differences are persistent in the vicinity of the proposed structure.

Eighth—We should carefully insulate all lead-sheathed cable from the building and from reinforcing metal. Generally it is sufficient to carry such cable on wooden blocks away from the building.

Ninth—Maintaining the reinforcing metal negative towards the concrete is not advisable, because of the results noted above, namely, the destruction of the bond.

Tenth—Watch the electrical insulation throughout your buildings wherever it is accessible, for here is a possible source of much trouble. Maintain it in good condition. In so far as is possible, place all wiring within metallic conduits.

Eleventh—Having inter-connected all metallic structures within the building, see that it is not grounded outside the insulating joints. The effect of ground-plates is local at best, and may give rise to passage of current from them to distant footings of the building and into the structure, at times giving rise to dangerous conditions.

For the complete data on this topic see Bulletin No. 18, Bureau of Standards, which also contains a comprehensive bibliography on the subject in general.

If, therefore, the above-named precautions are taken it is not likely that serious difficulties will be met, except in most pronounced cases, where special measures, not enumerated above, may be necessary.

Twelfth—Before taking up the second phase of our subject I wish to speak of another precautionary measure, namely, the constant co-operation desirable between those in charge of our buildings and those who transmit current through our streets, especially our electric railways, whose so-called "return currents" may be a source of danger unless properly cared for. By all means, keep in touch with railway men, to whom the matter is of direct financial concern, and who are anxious to place their current returns in as good condition as possible. If, therefore, you feel that at or near your buildings the rail bonding is imperfect or insufficient, do not lose time in getting into touch with the proper railway officials, whom you will find, as a rule, anxious to improve conditions of this sort.

I have taken the liberty to reproduce several illustrations from the above-named Government bulletin, showing graphically the effects of current flow on reinforced concrete.

ELECTROLYSIS UNDER GROUND

Because of ultimate economy in some cases, and of legislation in others, we are gradually placing all pipe lines and other means of transporting power, heat and cold, under ground. It is true of the refrigerating business in our larger cities that central plants are gradually displacing the smaller, isolated refrigerating plants. In market districts this is especially true. This has necessitated miles of brine piping beneath our streets. Needless to say, such lines, carrying as they do an excellent electrolyte, offer splendid opportunity for electrolytic corrosion.

Let us consider, then, the results of investigations made by the Government and by independent investigators upon this matter of corrosion of our underground metals.

Here we find both self-electrolysis and current-electrolysis more or less active. We find that the passage of current almost invariably stimulates self-electrolysis, and that in any consideration of the topic both must be considered.

The Government has confined its study to current-electrolysis, while independent investigators, notably the electric railways of St. Paul and Minneapolis and the Detroit United Lines, have considered both phases of the subject. Inasmuch as these investigators have in the main checked the results obtained by the Government concerning current-electrolysis, we will confine ourselves to the Government reports with regard to this phase, and to the results of the other investigators relative to self-electrolysis.

We will first consider current-electrolysis. What are the factors which must be taken into account in the study of this phase of corrosion of underground metals? They are:

First—Current density. This influences greatly the rate of corrosion. As a rule, it may be said that the efficiency of corrosion varies inversely with the current density, i. e., the greater the current density the less the efficiency of corrosion. It has been shown that in the case of soils well saturated with moisture the corrosion may vary from 20 per cent to 140 per cent between current densities of 5 milliamperes and 0.05 milliamperes per square inch, respectively. It is now believed that current densities lower than 0.05 milliamperes per square inch are not of any great consequence in practice, although they may be of considerable theoretical interest.

Second—Moisture present. This is possibly the most important factor next to current density. Moisture exerts a powerful influence upon the rate of corrosion with any current density. The point of maximum influence is that of soil-saturation, above which point increase in moisture does not seem to have appreciable effect.

Third—Oxygen present. By this we mean the accessibility of air to the system. The influence which oxygen bears upon corrosion has for long been a bone of contention among investigators. The Government says that in the case of iron immersed in a liquid electrolyte, oxygen has no appreciable effect.

Others, fully as well qualified to pass an opinion, show by experiment that oxygen does have a very appreciable effect under certain conditions. All investigators are agreed that oxygen has a marked influence upon the character of the end-products of corrosive action. Thus, if the rate of corrosion is rapid and the supply of oxygen small, there will be a greater amount of magnetic oxide formed, whereas if the rate is low and the amount of oxygen abundant ferric oxide will predominate. Bearing in mind, therefore, that around pipes in the ground the supply of oxygen is generally limited, we can, by a study of the oxides formed, arrive at some idea as to the causes of the corrosive action, having in the meantime taken into account all local conditions.

Fourth—Temperature. So far as we know at this time, the temperature of soils does not appreciably influence the rate of corrosion within the ordinary limits or range of temperatures found at the depths to which our structures extend, or at which our pipes are laid.

Fifth—Chemical and Physical Nature of Soils. Chemicals in soils exert some influence upon both the rate and the character of the resulting corrosion.

Thus, we find that nitrates and sulphates show an even corrosion with no pitting; carbonates somewhat less even, but still no pitting; hydroxides often show marked pitting, and, lastly, *chlorides* show very marked pitting.

None of the above-named chemicals produces the so-called "passive" condition with iron. Chromates seem to have some tendency in soils to produce passivity, but they do not cut down the rate of self-corrosion to anything like the same extent. Those chemicals which are known to inhibit corrosion of metals in liquids do not show this characteristic when the same metals are buried in soils. Pitting is effected, therefore, by both the character of the soil and by non-homogeneity of the metal in question.

Sixth—Depth below ground level. Depth of burial of pipes, etc., does not appear to influence the rate or character of corrosion.

Seventh—Several factors which may influence corrosion are: Probability of polarization and subsequent depolarization; possible formation of high-resistance films on the metal surface; the probability that in certain, as yet undefined, conditions pitting may actually retard corrosion and, lastly, the character of the metal itself.

The efficiency of corrosion in soils has little to do with the voltage found, excepting as it affects the current density, it having been demonstrated that corrosion is not appreciably different at voltages from 0.1 to 0.6 and from 5.0 to 10.0 volts. Experiments upon soils from widely separated districts indicate that corrosion efficiencies of from 50 per cent to 110 per cent may be expected in the case of iron buried in soils.

We have stated above that depth of burial as ordinarily found does not affect the rate of corrosion. It is necessary to modify this in this respect, namely, that if our structures are not below the frost line, we will find a difference in the soil resistance varying with the season. Thus at 18 degrees below zero it is something like 250 times that found in soil at 18 degrees above zero. It is evident, therefore, that soil surveys should be made in warm weather rather than in the winter.

In considering the possibility of corrosion from the return-circuits of electric railways we should consider average load conditions rather than peaks, for results of experimentation show that it is not the higher voltages which cause the most marked corrosion, since the rate of corrosion does not increase in anything like the same ratio as the voltage increase. For the full data, see Bulletin No. 25, Bureau of Standards.

SELF-CORROSION OF METALS IN SOILS

When two dissimilar metals, or two pieces of the same metals, are immersed in damp soils current is at once set up, which tends to flow without interruption, unless polarization sets in. It is known, however, that depolarization takes place quickly, and the current flow again begins, the flow being re-current for indefinite periods. The polarity of two pieces of the same metal will often reverse, but this is not true of two metals electrically dissimilar, such as copper and iron, where we always find the iron positive towards the copper. We should remember that current set up in this manner does not have to travel any great distance through the soil, which, if damp, offers a low resistance to the passage of current.

Experiments with mild steel immersed in soils for 66 days showed the following results, taking the losses in weight as a measure of the corrosion efficiency of the soils used: Clean sand showed the least action, although it may be pronounced if saturated with moisture; light, sandy soils, a little greater action; heavy clays carrying from 10 per cent to 15 per cent water next, and lastly, black, peaty soils showing the greatest action. In these soils the average rate of corrosion was 1.39 grams per year, at which rate an iron pipe of ¼-inch thickness would probably corrode through in about eight years. Under exaggerated soil and metal conditions the life may be even shorter.

A most interesting condition is found in the case of pipes placed

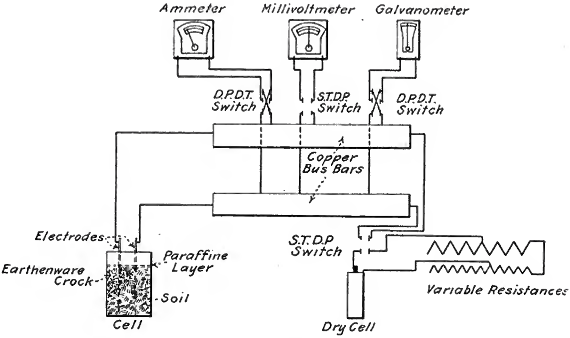


Fig. 5.—Typical Cell and Potentiometer Connections

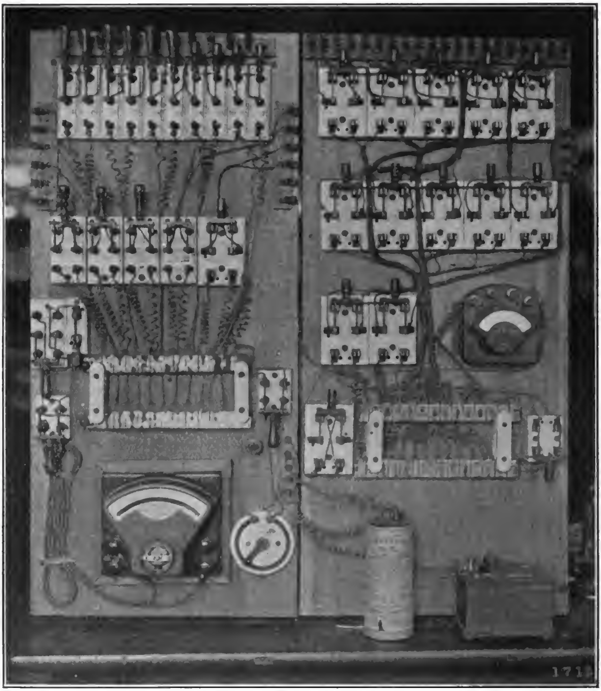
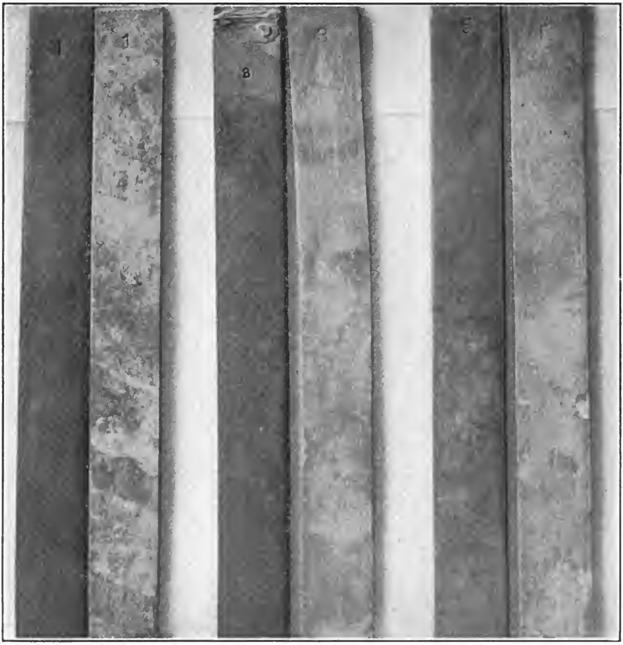


Fig. 6.—Potentiometer

in trenches and subsequently buried in a mixture of soils, or in two or more unlike soils in simple contact. Experiments show that iron corrodes faster under such soil conditions than it does in any one of the single soils, the most marked corrosion being shown at points of contact between two soils. Naturally two soils, to show this effect, must be in a state of unstable chemical condition, otherwise no difference of potential will be set up between them. When lead laterals in our streets are laid in trenches and buried in the excavated soils, we have just such a condition in many instances, and there is absolutely no doubt that much of the corrosion of such laterals is due to this very condition. There need be no current from outside sources to set up this action, but if present such current undoubtedly accelerates the action. The potentials found under such conditions are often as high as 1 volt where two dissimilar metals are immersed in a soil or where a single metal is immersed in two unlike soils.

Experiments carried out in Detroit with various soils in combination with copper, zinc, lead, cast iron and steel have shown very interesting results. Two unlike metals were placed in the soils with one end of each above the soil surface, these ends being

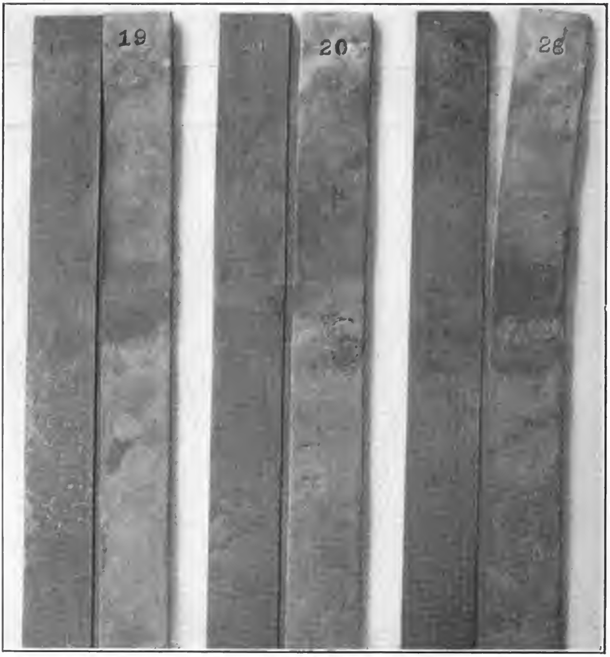


Copper-Zinc Electrodes.—Strips of copper and zinc immersed in a sandy-clay soil for three months. The metals were not in metallic contact in the soil. They were connected through an external circuit to a potentiometer, by which the voltage and the amperage were determined. The whole apparatus was thoroughly insulated and no external electric current could enter the system. Because of the difference in solution pressures of the two metals in the soil, which acted as an electrolyte, a current was set up between the two metals, zinc being the anode of the cell. The anode was rapidly attacked under these conditions.

Fig. 7.—Metals in a Sandy Clay Soil

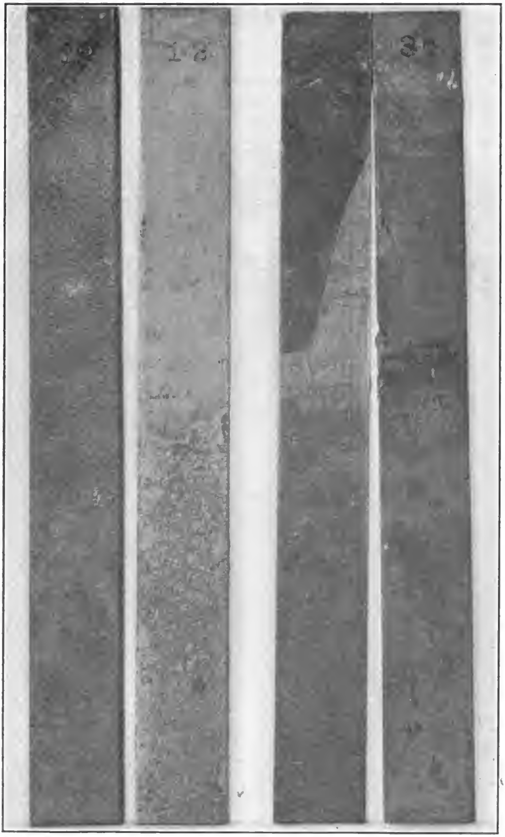
connected through a sensitive potentiometer, by which it was possible to read the voltage and the amperage of each such cell, Figs. 5 and 6.

Thus, copper-zinc, lead-steel, steel-cast iron and lead-cast iron electrodes were placed in the various soils and left for three months. In this time the characteristics of each cell were noted and the losses in weight also determined. Sixty such cells were set up. The soils were placed in earthenware crocks, the electrodes inserted



Lead-Cast Iron Electrodes.—Strips of lead and cast iron immersed in a clay soil for three months. Metals not in contact in the soil. Metals connected to a potentiometer through an external circuit, whereby the voltage and amperage were determined. The entire system was well insulated against the entry of external electric current. In this case both the metals were attacked, the polarity of the system varying during the test. The cast iron corroded evenly, whereas the lead is pitted badly in local areas.

Fig. 8.—Metals in Clay Soil



Copper Zinc C. Iron Lead

Cell No. 12 Cell No. 30

Fig. 9.—Auto-Electrolysis of Metals in Sandy Loam

and the surface of the soil covered with ½ inch of paraffin to prevent loss of moisture. In all cases the moisture content was fixed at 30 per cent by weight of the dry soils. Needless to say, this percentage of moisture did not give the same apparent wetness to the various soils, but it was necessary to have a standard moisture content, and experimentation showed that 30 per cent of moisture with average soils should give about 75 per cent corrosion efficiency. The loss in moisture from cells protected with paraffin was found to be less than 0.25 per cent in three months' time, an amount so small as to be negligible. Figs. 7, 8, 9 and 10.

In soils containing considerable salt the current set up may be as high as 50 milliamperes at a potential of 600 millivolts, a quantity

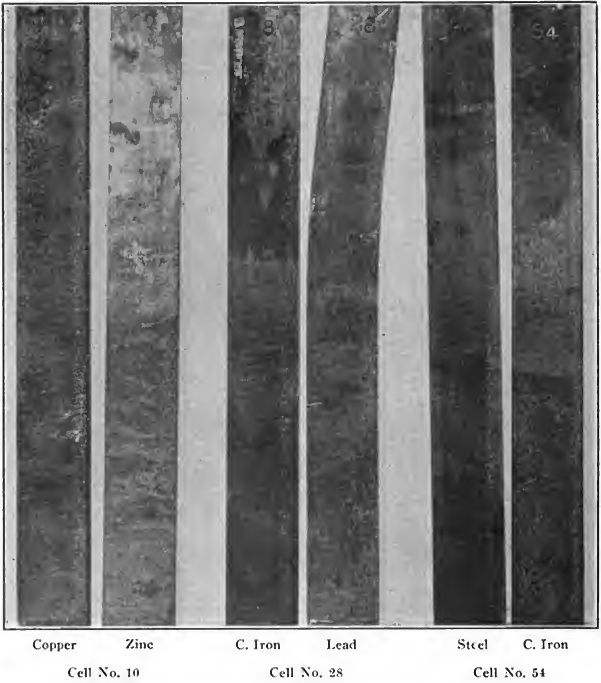


Fig. 10.—Auto-Electrolysis of Metals in Wet Sand

sufficient to cause very marked corrosion of the anode, especially with a cell having copper-zinc electrodes, where the solution pressures of the two metals are widely different.

Observations of the corrosions of pipe laterals in the streets of Detroit seemed to indicate that there must be factors at work other than the action of stray electric currents, for such corrosion was noted in many instances where absolutely no current flow was detected with very sensitive meters. Analyses of the soils in such cases indicated the presence in concentrated form of street refuse, which had found its way by seepage between the pavement and curbing down to the piping beneath the surface.

Cells set up with street refuse containing 30 per cent moisture proved that such material will act markedly as an electrolyte, and, also, such material is generally corrosive in action from a purely chemical standpoint. Lead immersed in such material loses weight rapidly, the action being characterized by marked pitting. Here we have corrosion going on in the absence of any electric current, and

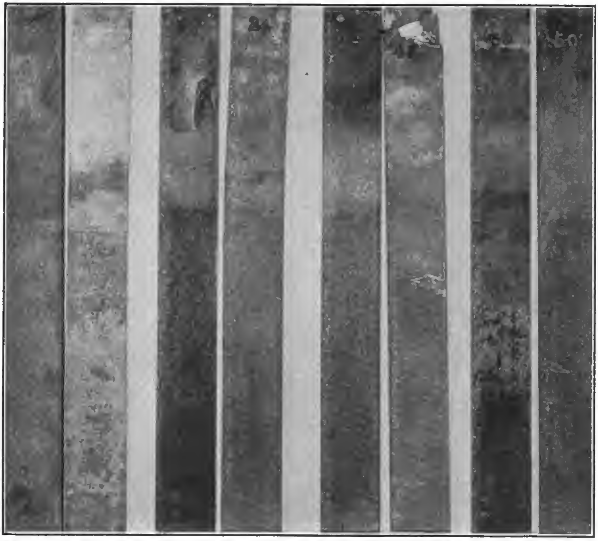
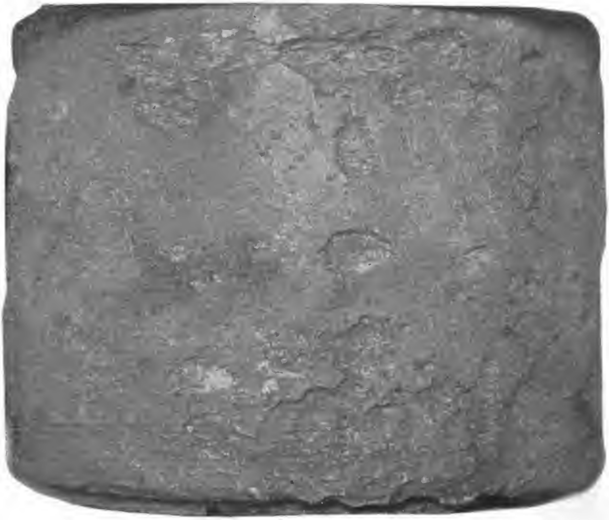


Fig. 11.—Auto-Electrolysis of Metals in Street Refuse

there can be no doubt whatever that much of the corrosion of lead laterals beneath our streets is caused by soil action or by street refuse. Fig. 11.

There is still another dangerous condition of piping which we cannot pass by without comment, namely, the burial of pipes in cinders. A case has recently come to my attention where iron pipe was rendered useless through pitting under such conditions. When this pipe was taken up it was found to be heavily coated with cinders, and beneath the particles of carbon (cinders) pits were invariably forming. This pipe carried brine, and when the pipe was once punctured the corrosion was undoubtedly accelerated through the presence of brine around the pipe. Before taking up this pipe potential and current measurements were taken, there being only a small potential drop along its length, and no current flow which could be detected with a milli-ammeter reading to 0.005 milli-ampere. This pipeline lasted about six years under these conditions. The corrosion in this case was undoubtedly due to self-electrolysis



Wrought iron coupling, 30 inches in diameter, laid in a cinder-fill in a region where there could be no stray electric currents. Note the deep pitting and also general corrosion due to the presence of carbon in the cinders in the presence of water.

Fig. 12.—Self-Electrolysis

set up between the carbon of the cinders and the iron of the pipe, moisture and, finally, brine, each contributing to the rate of corrosion. Fig. 12.

I may state finally that it will do little good to insulate our piping against heat and cold unless we see to it that such insulating material cannot become saturated with moisture, for in a saturated condition they will serve admirably as conductors of current.

I realize that to many of you data brought out may be only of slight interest; in fact, to some it may seem somewhat far-fetched to assume it to be of any real value to refrigerating engineers. I trust, however, that when erecting concrete structures, or when placing metallic structures beneath the ground, you may find some hints herein which will better enable you to conserve them against the ever-present danger of corrosion.

DISCUSSION

F. L. Fairbanks—I would like to ask Mr. Smith one question. We have a steel and concrete warehouse in which we have been getting a great deal of corrosion at the column footings, and also on our brine mains just outside of and where they enter the warehouse. Our tests show that the direction of current, as well as the difference in potential, vary at different times; at first being into the warehouse and then out of the warehouse, and I would like to ask if he has any idea of what may be the cause of this condition. The difference in potential varies from, say, 18 to 75 milli volts.

Mr. Smith—Have you noticed any difference during the day? Is it higher in the morning than the evening?

F. L. Fairbanks—No, the change is during the day.

Mr. Smith—Without any special regularity?

F. L. Fairbanks—Yes and no; the only regularity we have been able to determine is that it seems to be synchronous with the tides. It reverses itself twice daily, as shown by readings taken regularly for several days.

Mr. Smith—Possibly, I can throw a little light on that; I don't know that it fits the case, but at St. Claire, Mich., is located a large salt works, and we have considerable trouble there where one of our interurban lines parallels some of their brine lines. We made extended surveys of the soil and conditions of potential and current flow between our rails and their pipes, and also between the pipes and the apparatus within their buildings. We found that at times they were responsible for the current on the pipes, and at other times we were responsible. When our cars are running parallel to their plant, especially when several cars would go through, perhaps a train of four cars, stopping and starting at a bridge nearby, the accelerating current is very high, and at those times their pipe lines received considerable current from our rails, but when our cars were not nearer than one-half a mile in either direction, then the railway apparently had no effect on them whatever. The current was emptying from their plant to the pipe line constantly. Your case may be something on that order.

F. L. Fairbanks—We assume, of course, that there is a bare possibility of current going and coming from an elevated railroad structure, although this is some distance away. There are no cars on the street, no rails, and our own current is alternating, which would eliminate this as the probable source. Our tests show direct current, which reverses its polarity with the tide.

Mr. Smith—There is one phase of the study of the effect of electric current we are investigating at the present time, which is entirely apart from refrigeration, that shows how far the effect of electrical current may be felt, namely, the effect of electric current upon trees. It so happens that, when an electric railway secures a right of way through the country, it has to guarantee the preservation of the trees as far as possible. We had a case not long ago where the line was sued for quite a large sum of money for the death of a tree situated over 1,600 feet from our rails, and we thought it was absolutely absurd and did not hesitate to say so. When we came to make surveys, we found there was a marked difference of potential between that point and our rails and, by setting apart the 1,600 feet into lengths of 40 feet, we found there was a fairly uniform drop in potential. That tree was unquestionably getting current through its roots, and I believe that it actually killed the tree; we paid for it, anyway. I really do not know, Mr. Fairbanks, to what to ascribe your trouble, unless it is caused by alternate wetting and drying of the walls of the building when exposed to tidal action. A waterproof covering ought to stop this effect.

F. L. Fairbanks—It is a serious matter with us, as we have lost 30,000 or 40,000 gallons of brine due to electrolysis of the pipe lines, and heavy copper cables with heavy copper plates as collectors seem to have no appreciable effect. I believe that the only sure way to protect a line is by insulation; insulation will stop it if carefully done.

Mr. Smith.—I was going to ask if you had put in insulating joints wherever pipes enter or leave your buildings?

F. L. Fairbanks—Not only did we insulate the joints, but we insulated the pipe line itself.

Mr. Smith—You have been able to cut it down that way?

F. L. Fairbanks—We have been able to stop it as far as the pipe line is concerned, but we get it at some other point.

Mr. Smith—I could not state the cause without knowing all the conditions. In the experience I have had in the study of corrosion, I have yet to find two cases alike.

Bruce Walter—I have a few notes on an experiment I made on corrosion of pipes in an ammonia condenser, that I think will be of interest to the Society:

In the year 1904 the air refrigerating plant at the Isabella Furnaces of the United States Steel Corporation was completed. At that time there was considerable difference of opinion as to the relative durability of steel and iron pipe, so following the older practice we decided to use wrought iron throughout. In 1905, in

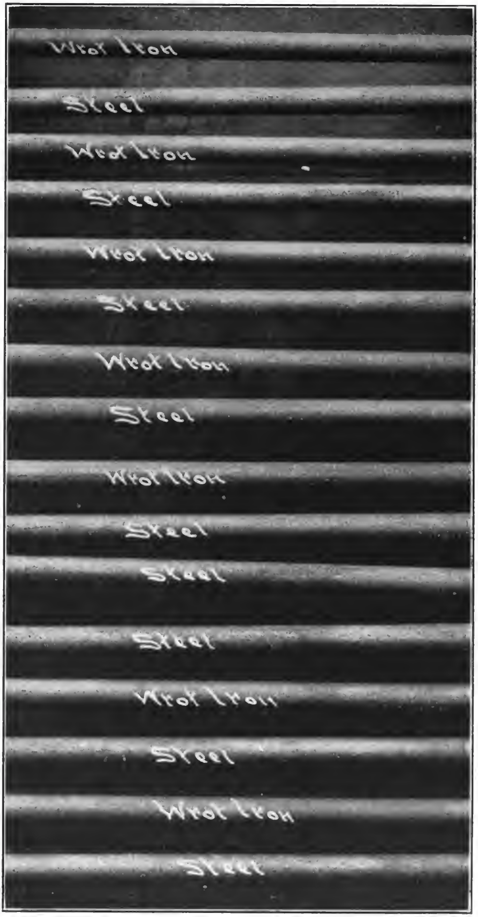


Fig. 13.—Pipe Installed December 4, 1905, and Removed September 1, 1915

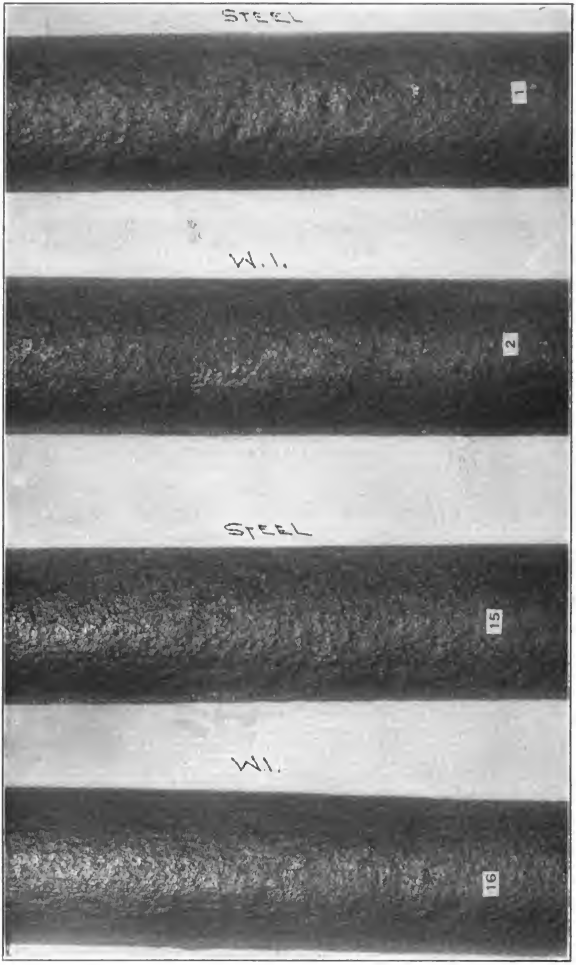


Fig. 14.—An Enlargement of the Two Top and Bottom Pipes Shown in Fig. 13

order to determine the question of durability, part of one coil of the ammonia condenser was dismantled and rebuilt of alternate lengths of standard weight wrought iron and steel pipe. A few months ago it seemed advisable, on account of corrosion in certain parts of the system, to renew these condenser coils.

A close surface inspection of all this pipe was made after the system was dismantled. No corrosion whatever was found inside. Every length showed considerable corrosion outside, there being no appreciable difference between the various pipes so far as could be detected by the eye. No leaks had developed in any of the test pipes. Pieces about 6 feet long were then cut from each of the pipes in the test coil and numbered in rotation from the top of the coil downwards. The rust was cleaned off with a wire brush, and measurements made of the depth of several of the deepest pits in each length. Each piece was also analyzed and tested to check up the records as to the identity of the material. The relative position of these pipes in the coil and the appearance after being cleaned from rust are shown in Figs. 13 and 14.

All these pipes had lost about one-third of their average thickness. Measurements of the pits in each length from this test coil resulted as follows:

	Steel	*Iron*
Average depth of deepest pit	.048 inch	.045 inch
Average depth of ten deepest pits	.037 "	.037 "
Depth of deepest pit found	.055 "	.054 "

The average outside diameter of both the wrought iron and steel pipe after cleaning, as figured from circumferential measurements, was found to be 2.34 inches, whereas the standard outside diameter for this size pipe is 2.375 inches, showing a loss of .035 inch due to corrosion. Adding half this general loss to the depth of pitting gives the maximum depth of corrosion.

A similar examination of the wrought iron pipes used in the other coils of this system showed them to be in the same condition. In short, there was clearly no difference in corrosion between the wrought iron and steel pipe after practically ten years' actual service conditions.

N. H. Hiller—Did you weigh the pipe?

Bruce Walter—No, we did not weigh the pipe.

*Mr. Smith**—Mr. Fairbanks' case undoubtedly illustrates the fact that the causes of corrosive action are often far removed from the point where corrosion is manifested. I am inclined to believe that through waterproofing of both the outer and inner walls of the building, where exposed to alternate wetting and drying, should greatly cut down the entrance of electric current into the building and also into apparatus situated therein.

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* Author's closure under the rules.

THE WORK OF THE BUREAU OF STANDARDS ON CONSTANTS OF REFRIGERATION

By H. C. Dickinson, Bureau of Standards, Washington, D. C.

I have a special purpose in presenting this review of the work that the Bureau of Standards has been doing, namely, that this Society may be informed of the progress of the several investigations along this line. You understand that the program which has been undertaken was considered more or less carefully by a committee of this Society and members of the Bureau staff, but I think it would be well for us all to keep in closer touch with what is being done. I shall try to cover as briefly as possible a number of points and shall be pleased to answer at any time any questions which may arise. In order to cover the ground, extreme brevity will be necessary. With reference to the papers presented tonight on "The Testing of Thermal Insulators" and "Preliminary Measurements of the Specific Volume of Anhydrous Ammonia," the latter based on the work by Dr. Harper and Mr. Cragoe, and the former, the work on conductivity, most of which has been done by Mr. Van Dusen, aggregate approximately 25 per cent of all the work done by the Bureau on constants of refrigeration within the time covered by these papers. There are a number of other problems, some of which have occupied much more time than either of these investigations. These I shall try to outline very briefly.

Dr. Harper has explained quite clearly why we do not wish to be premature in presenting results in numerical form. It seems important to present only the final numerical results, avoiding the publication of figures which may need to be changed even slightly at a later date.

There are a number of separate topics to be presented, and if there are any questions on any point do not hesitate to bring them up immediately. The first piece of work to be presented is the study of the specific heat and latent heat of anhydrous ammonia. The latent heat appears in Clapeyron's equation, and practically no observations of real value have been made on this constant since the time of Regnault. His twelve experiments represent perhaps the most important published experimental work on this subject. The Bureau's results, as yet unpublished, are almost certainly good within 0.5 per cent for the whole range, but much better results than this are to be expected before publication. Fig. 1 represents the aneroid calorimeter which was shown you last year in diagrammatic form as it was used for determining the specific heat and latent heat of fusion of ice. It consists essentially of a copper shell fitted with an electric heating coil and a resistance thermometer, the copper shell containing a steel cylinder in which can be placed water, ice or, in this case, liquid ammonia. Last year this calorimeter was used with ice and water; latterly with liquid ammonia by adding

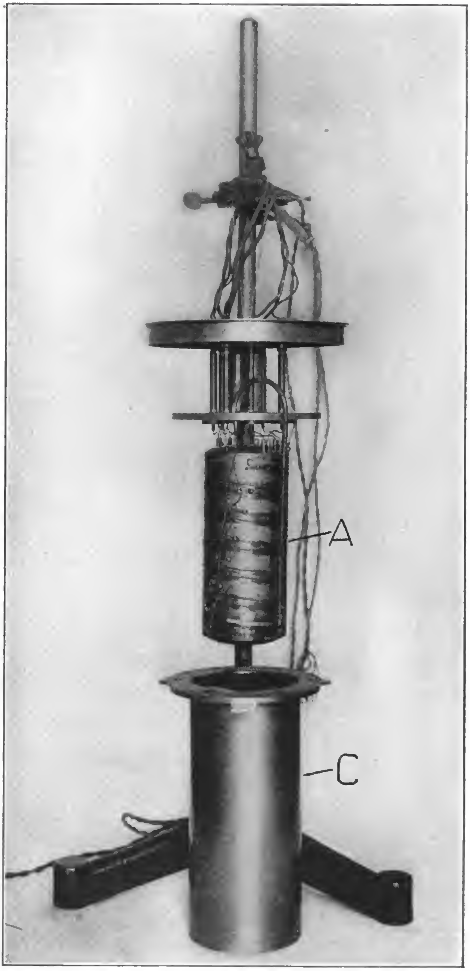

Fig. 1.—Aneroid Calorimeter; Detail View

thin steel tubes for filling and emptying the cylinder. The cylinder, *A*, is provided with cross vanes to equalize the temperature.

The specific heat of liquid ammonia is determined by filling the cylinder as full of ammonia as the conditions will permit (the whole container, *C*, being immersed in a bath, the temperature of which is thermostatically controlled), raising the temperature of the cylinder, measuring the amount of energy supplied electrically, while keeping the temperature of the outer shell continually equal to that of the cylinder, and measuring in that way the heat capacity of the whole system. Knowing the heat capacity of the metal parts, the heat capacity of the ammonia can be determined.

The latent heat is determined by allowing a small amount of ammonia to evaporate from this container, flowing through the fine capillary and being absorbed or rather condensed in another steel container at a lower temperature. The procedure is simple; with the temperature constant, a valve is opened permitting ammonia to evaporate slowly, and at the same time enough heat is supplied electrically to keep the temperature constant, i. e., to balance the latent heat of vaporization of the liquid which evaporates. At the end of the experiment the ammonia valve is closed, the heat is also shut off by opening a switch and the whole is allowed to come back to equilibrium as nearly as possible at the original temperature. The temperature and the amount of heat supplied are measured, as well as the amount of ammonia evaporated; thus the latent heat is computed, numerous minor corrections of course being necessary. There are other details which I will not go into.

Fig. 2 shows the corner of the laboratory in which this apparatus is mounted. The potentiometer, *P*, serves to measure the amount of energy supplied to the heating coil. The Wheatstone bridge, *W*, serves to measure the temperature by means of platinum resistance thermometers with a sensitivity of about a ten-thousandth of a degree. The figure also shows two galvanometers mounted on the walls, telescopes for observing the same, and the pressure gages which indicate the pressures in the carbon-dioxide refrigerating system used for controlling the temperature of the bath in which the calorimeter is immersed. The calorimeter is shown standing outside the bath, with its jacket removed.

The results with this apparatus have been fairly satisfactory, giving an accuracy of possibly 0.3 per cent. Considering the fact that not only ammonia, but all the other commercial refrigerants, as well as some other substances, were to be studied by the same method, it seemed that considerable improvement in convenience and in accuracy could be profitably obtained by re-designing and rebuilding the apparatus; therefore considerable time has been spent in building not a duplicate but a successor to the apparatus here represented. The parts of this new calorimeter are shown in Fig. 3. *S* is a steel shell, which has been tested to 2,000 pounds pressure per square inch. The two steel ends, *e*, *e*, screw onto the ends of the shell, and like all the remainder of the inside are carefully coated with pure tin, protecting them from rusting and from the action of the am-

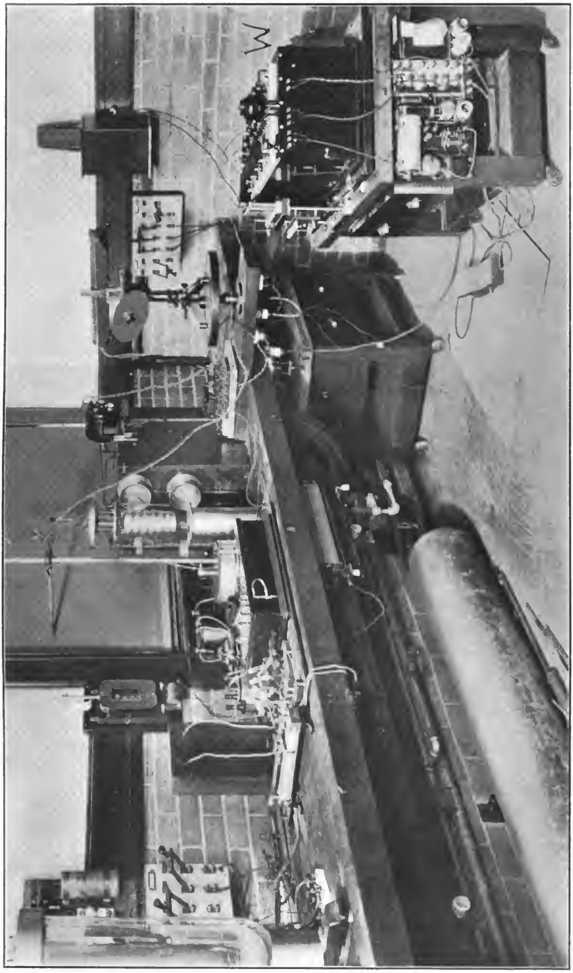

Fig. 2.—Aneroid Calorimeter and Accessories. For Measurement of Specific Heats and Latent Heats to any Temperature, − 50° to + 200° C.

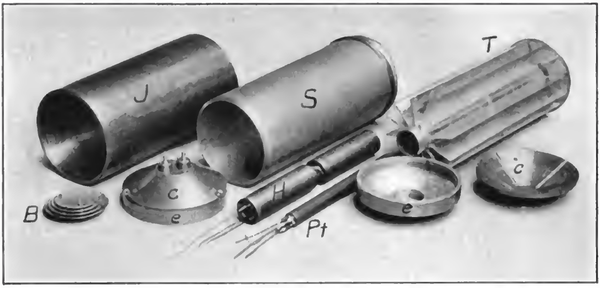

Fig. 3.—Latent-Heat Calorimeter of Aneroid Type—Disassembled

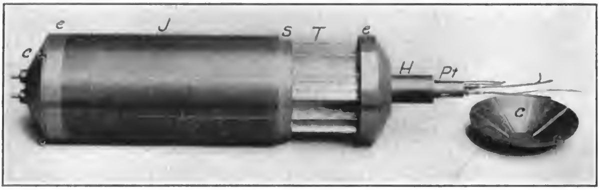

Fig. 4.—Latent-Heat Calorimeter of Aneroid Type—Partially Assembled

monia. A copper cylinder or sleeve, *J*, slips over the steel shell, with a very narrow air space between. Copper caps, *c*, *c*, fit over the steel ends, *e*, *e*, also with a narrow air space except at the outer edge. *H* represents a double electric heating coil, which fits exactly inside the tube *T*. The two parts of this heater may be used independently, so as to distribute heat on the inside of the calorimeter with reference to the level of the evaporating liquid. If the level of the liquid is toward the bottom, heat is supplied at the bottom, and conversely. A standard type of platinum resistance thermometer is built within a brass tube, *Pt*, which fits inside the heating coil. The tube *T*, into which heater and thermometer are fitted, carries a number of tin-covered steel vanes, which serve to quickly equalize the temperature throughout the liquid in the calorimeter. The thermometer in the former calorimeter previously described was not entirely satisfactory, since the method of construction imposed certain limitations. In the new calorimeter we have an excellent thermometer, which will certainly be reliable to a few ten thousandths of a degree. *B* is a system of baffle plates, which are placed just below the outlet tubes to prevent spray from passing out with the vapor. The presence of particles of liquid in the vapor is one of the chief causes of error in most measurements of heat of vaporization, and this system of baffles, as well as the system of temperature distribution, has been carefully designed to avoid the possibility of such errors. Fig. 4 represents the same apparatus partly assembled. The copper sleeve is in place while the other parts are ready to assemble.

The reason for the copper shell being separated from the steel cylinder by a narrow air space is as follows: At some point in the cylinder, when used for latent heat measurements, there must be a free surface of liquid where most of the cooling is taking place. The method of measurement requires that the temperature all over the surface of the calorimeter be known, and this copper shell, which is not in contact with the steel surface, tends to smooth out the temperature gradient and increase the accuracy of the temperature measurement.

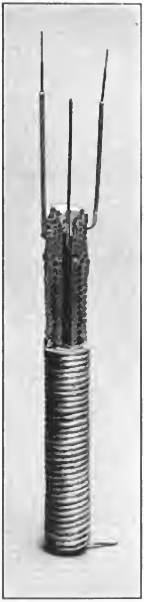

Fig. 5.—Flow Calorimeter for Latent Heat Measurements. Spiral Tube to Carry the Refrigerant, Surrounded Inside and Outside by Spiral Tubes Containing Electric Heater Wires

Another entirely different method is also being used for the determination of latent heat. Fig. 5 is a cross section of a discarded calorimeter like the one which is being used. At the center is a small tube, serving as an inlet for liquid ammonia. This tube ends in a platinum capillary, and the liquid ammonia enters through the capillary into the inner space whence it flows back, entering the coil, flowing the length of this coil and out into another coil which is separated from this, but is not shown in the figure, and on to the outside. Inside and outside of this coil is wound a copper tube containing an electric heating element which serves to supply accurately enough heat to compensate for the cooling effect of the evaporating ammonia. We have here liquid ammonia entering and evaporating and the vapor leaving at the same temperature; thus this is virtually a refrigerating system in a very small space in which heat can be supplied exactly as required to compensate for the cooling. This method is thus entirely different from the previous method. It has given considerable trouble in some respects which were unforeseen, but it bids fair to give eventually excellent results as a check method.

The Secretary—How do you check one against the other?

Mr. Dickinson—As far as results have been obtained, we check within the limits of accuracy of those results. There are as yet only a few observations by the latter method which are reliable.

The Secretary—What per cent?

Mr. Dickinson—Not over 0.5 per cent, but it is too early to compare results, since we have not yet obtained sufficient observations.

Fig. 6 represents a low temperature bath. This bath was shown in a previous paper by Dr. Harper, as it appears at present enclosed in a metallic jacket. The original glass flask was destroyed by an explosion of one of the ammonia bulbs. The explosion scattered gasoline, Dewar flask, and other things all over the room. The apparatus was then rebuilt with a metallic shell to protect the observers.

This apparatus was originally used with the flow calorimeter just described, but a new low temperature bath has been built which takes its place, and this is now being used, as explained by Dr. Harper. Fig. 7 represents the apparatus now being used with the flow calorimeter. It is similar to the one shown in Fig. 2. The temperature can be regulated between — 50 C. and + 50 C. to an accuracy of two thousandths of a degree at most points. The auxiliary apparatus, potentiometer, etc., are shown as in Fig. 2.

The work with the aneroid calorimeter has been carried on by Mr. Osborne with the assistance of Messrs. Van Dusen and Meyers. The work with the flow calorimeter has been carried on by Mr. James.

A supplementary investigation which is in progress under the supervision of Mr. Mueller, not directly under refrigeration constants, concerns an accurate determination of the vapor pressure of ammonia over a rather limited range of temperature; perhaps the

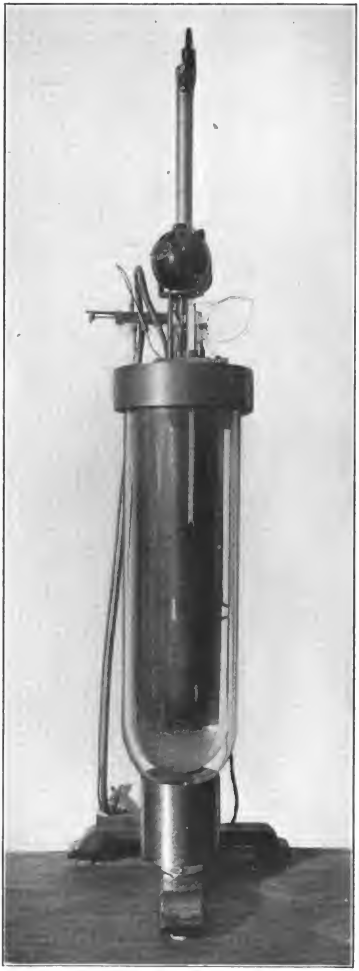

Fig. 6.—Low-Temperature Thermostat Bath. Dewar Vacuum Flask with Stirrer, Electric Heater, Refrigerating Coil, and Temperature Regulator

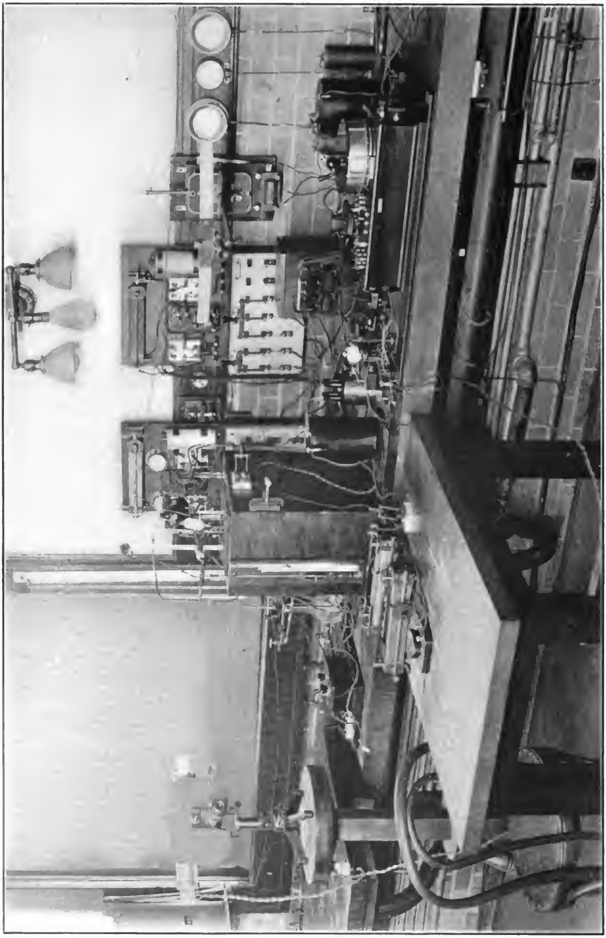

Fig. 7.—Flow Calorimeter and all Accessories. The Central Unit, at the Center of the Wooden Box, is the Calorimeter Shown in Detail in Fig. 5

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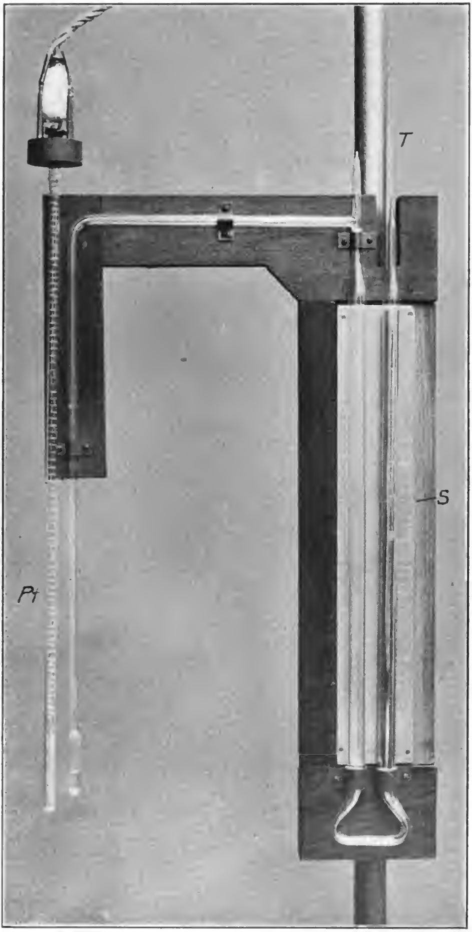


Fig. 8.—Apparatus for Measuring the Boiling Point of Ammonia. The Boiling Point of Pure Anhydrous Ammonia is about — 33.3° C. or — 28° F.

most precise measurements ever made of this constant at temperatures near its normal boiling point. The apparatus shown complete in Fig. 8 has been used for this purpose. There is a small glass bulb connected through a glass capillary to a U-tube containing mercury. This apparatus is useful as a constant temperature indicator as well as for measuring vapor pressure. Any volatile liquid, even liquid air, can be placed in a bulb of this type, the open end of the mercury tube sealed and the apparatus used at any time afterwards by immersing the bulb in a bath at the proper temperature (within a few degrees of the normal boiling point of the liquid in the bulb) to be measured, breaking off the glass tip shown at *T*, allowing the atmospheric pressure to act upon the surface of the mercury, and measuring on the scale the excess or deficiency of pressure compared with the atmosphere. The platinum resistance thermometer, *Pt.*, is mounted alongside the vapor bulb to measure the temperature of the liquid in which it is immersed. In the present instance ammonia was distilled into the bulb, and vapor pressures near the boiling point have been determined with considerable precision. Fig. 9 represents some of the results obtained. The lower curve covers the range from — 40 degrees C. to — 28 degrees C. The upper curve represents an enlargement of the lower one; that is, covering only the range near the boiling point from — 33.9 to — 32.9. The scale is rather open; for instance, the range 745 to 750 on the upper curve represents five parts in seven hundred, *i. e.*, a little less than 1 per cent for a whole division. You will notice how closely the various points fall on the curve. One point

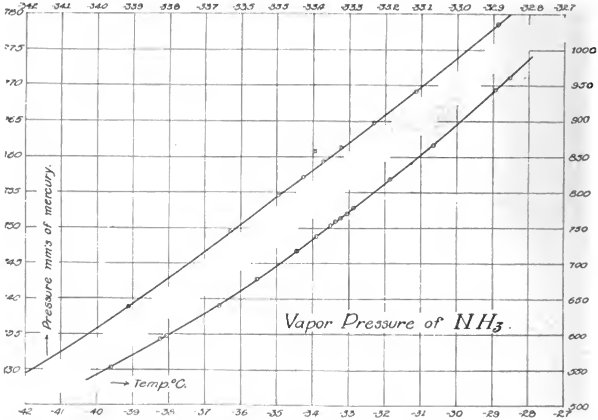


Fig. 9.—Vapor Pressure of Anhydrous Ammonia. The Upper Curve is the Central Portion (about One-Tenth) of the Lower Curve Much Magnified

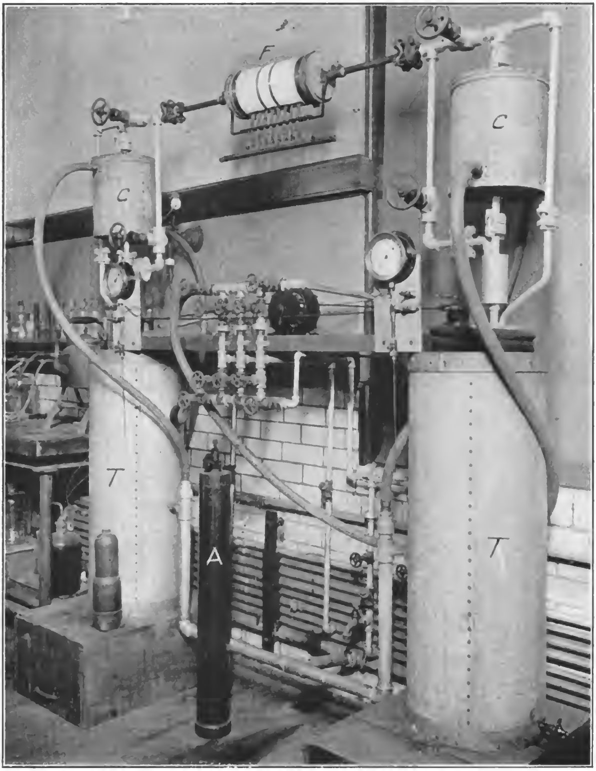


Fig. 10.—Experimental Absorption Refrigerating Plant for the Study of the Formation of Non-Condensible Gases in Such Plants

is somewhat off the curve, but there is a definite reason for that. This is an interesting piece of work which was not originally considered as an essential part of the program.

Fig. 10 represents a piece of appartus which was built in connection with the chemical part of our investigation. This was referred to this afternoon as the apparatus in which Mr. McKelvey is hoping to make a careful analytical study of the formation of non-condensible gases, etc., in ammonia systems. I shall outline briefly how this apparatus is built. The two containers, tanks *T T*, are alike; they are oil baths, in each of which is immersed an ammonia cylinder like the one shown at *A*. The ammonia cylinders on the two sides are connected through a trap, two condenser coils, *C* and *C*, and a furnace, *F*, at the top, with the necessary valves, pressure gages, by-passes, etc., for regulation. The two containers or tanks are so arranged that the liquid surrounding the ammonia cylinders can be interchanged between them; for instance, if we have hot oil in one and cold oil in the other, by means of a pump the cold oil can be passed over to the warm tank, and simultaneously the hot oil to the cold tank.

The method of operation is somewhat as follows: Strong aqua ammonia is placed in one of the cylinders and water in the other, or perhaps less concentrated aqua ammonia in both, and the temperature of one of the cylinders is raised and that of the other lowered by means of the heating and cooling coils provided inside the tanks. We have in effect a generator, a rectifier and a condenser (one of the condenser coils acting as a rectifier), by means of which we can pass ammonia vapor at any desired pressure from the generator through the furnace, *F*. The temperature in this furnace can be controlled, and the ammonia at the desired pressure, within moderate limits, can be passed through at any rate desired, also within moderate limits. The effect, therefore, of passing ammonia from an absorption system through piping under varying conditions of pressure and temperature, in the presence of various reagents, placed in the furnace, can be studied. The operation can be repeated as many times as desired, since when a sufficient amount of ammonia is passed over in one direction the cold medium from the tank on the absorbing side is pumped over to the generator side and the hot liquid from the generator side to the absorber side, and the whole process repeated, the vapor passing in the opposite direction. Thus the process can be carried on as long as desired, when the charge can be tested for production of foreign gases, etc. The method of simulating the compression system is evident. Of course, there is no means of reproducing the action of the compression cylinder; we can simply introduce liquid ammonia into one cylinder and pass the vapor alternately from one cylinder to the other through the furnace, which may contain any materials suspected of having an effect on ammonia under working conditions in compression plants. If that is clear, I will pass on to the next subject.

The Secretary—Do you secure the ammonia you are testing directly from the ammonia manufacturer, or is it some ammonia that has been taken out of a refrigerating system, ammonia that has been used?

Mr. Dickinson—Eventually it is intended to test all sorts of ammonia. I think probably at present the method will be to add known impurities to pure ammonia, rather than try to work with ammonia taken from a system in operation, and which contains indeterminate amounts of unknown impurities.

The Secretary—It is these unknown impurities that cause so much trouble in the refrigerating system.

Mr. Dickinson—Would results from a number of samples of that kind mean anything? Would it be possible to know that they are like any other samples one may encounter?

The Secretary—That may be true, but we want to know what are the impurities that are in the ammonia that has been through our refrigerating system.

Mr. Dickinson—It is possible to determine with some measure of certainty what impurities are likely to be present in a working system, and by introducing those impurities, one at a time, it may be possible to find which, if any, are producing the troublesome effects.

F. W. Frerichs—May I ask whether these cylinders are coated inside or are pure iron?

Mr. Dickinson—They are pure iron. The inner surfaces have all been carefully cleaned or turned to remove all scale and surface impurities.

F. W. Frerichs—Have you got the analysis of that iron?

Mr. Dickinson—I believe samples of it have been taken; whether it has yet been analyzed I am not sure.

Just another word as regards the chemical work. Ten samples of ammonia have been obtained, commercial samples, three of which have come from Germany. These samples have been tested for non-condensing gases in the liquid and vapor phase, for ammonium acetate and for water remaining after evaporation at both low and high pressures. A method has also been perfected for determining the carbonaceous material in the ammonia by burning a sample of it mixed with oxygen and determining the carbonic acid formed.

Some physical constants have been tested as criteria of purity. The variation of the boiling point or the vapor pressure which I have just described does not serve as a satisfactory criterion of purity, so far as it has been tried, as the presence of air is of prime importance, masking the effect of other substances. The density and electrical conductivity have been tried out to a very limited extent.

Fig. 11 represents apparatus used for determining the freezing point of samples of ammonia with a view to using this constant as a criterion of purity. Two samples of commercial ammonia, one of which was supposed to be as pure as any on hand, the other of which was known to be rather impure, gave a difference in the freezing point of about one-half degree C., so that the freezing point would not serve as a convenient commercial criterion of purity, as it requires rather careful work to measure the freezing point with sufficient accuracy to indicate the presence of small impurities. However, in a laboratory where precise temperature measurements can be made, this criterion may be extremely valuable.

The apparatus shown in Fig. 11 contains a platinum resistance thermometer which may be seen inside the glass vessel. The glass tube which contains the ammonia is surrounded by a glass jacket resembling a vacuum flask, the jacket of which can be evacuated

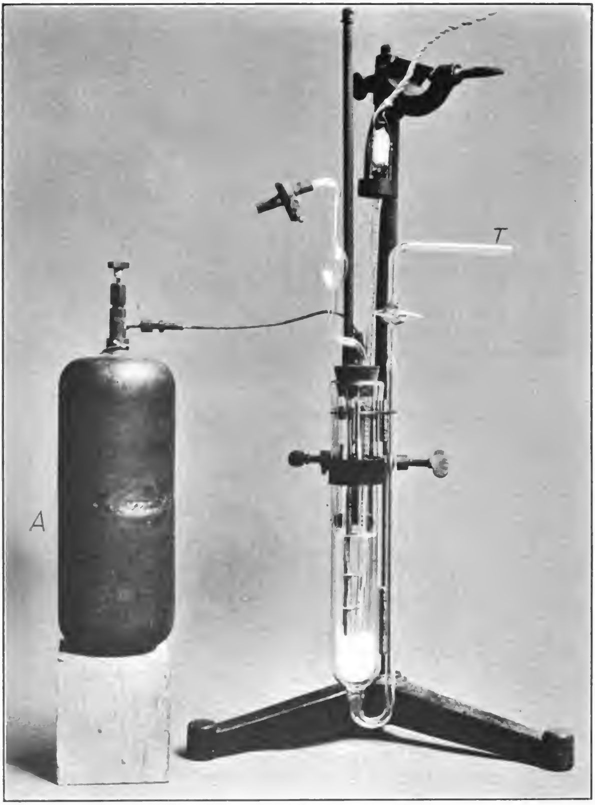


Fig. 11.—Apparatus for Measuring the Freezing Point of Ammonia. The Freezing Point of Pure Anhydrous Ammonia is about —77.7° C. or —108° F.

when desired by pumping air out through the tube *T* shown at the right of the figure. One of the small ammonia containers that was used for filling is shown at *A*. The method is to draw liquid ammonia into the freezing tube, and cool down first as far as possible by any convenient means, then again fill the tube with ammonia to supply what has evaporated, exhaust with an aspirator and let the ammonia boil away, cooling down to near its freezing point; then connect to sulphuric acid flask and lower the pressure to a point where the ammonia freezes by its own evaporation. This proves to be a convenient method of obtaining solid ammonia. If a supply of liquid air is available the ammonia may be frozen by immersing the container in the liquid air. It crystallizes into large, solid crystals, which tend to freeze together and cannot readily be stirred. The freezing point of the purest samples used was found to be — 77.7 degrees C., which is — 107.9 degrees F.

F. W. Frerichs—I have not quite understood in regard to the boiling point; has any difference been found between pure and less pure ammonia?

Mr. Dickinson—The boiling point is probably affected so much more by the presence of non-condensible gases, air, etc., than by other impurities, that there is great difficulty in obtaining a boiling point sample sufficiently free from air. It is evident that a commercial sample which contains more or less air would not serve for boiling point determinations; the main result obtained would be only some indication as to the amount of air the sample contained. This in itself is not of prime importance.

F. W. Frerichs—How many degrees, or what part of a degree, can the content of more or less air make in change of the boiling point?

Mr. Dickinson—I could not answer that question off-hand. Though the effect of the amount of air often present would be very considerable, I would not dare to make a statement as to the figure.

Some two years ago the Bureau issued a circular on the freezing point and heat capacity of various concentrations of calcium chloride brine. There have been many inquiries as to the same constant for sodium chloride of commercial purity, and as a supplementary investigation, the Bureau is now determining the heat capacities and freezing points of sodium chloride brines with the same apparatus which we described some time ago in the Bulletin of the Bureau. This apparatus, shown in Fig. 12, consists of a large Dewar flask in which a given amount of brine is placed, and the heat capacity determined by supplying a measured amount of heat electrically and measuring the rise of temperature. This work is in progress, and in connection with it determinations of the change of density with temperature are to be carried out.

Fig. 13 represents the carbon dioxide compressor which supplies the cooling for all the low-temperature calorimeters. This machine is now giving satisfactory results, except that we are not good enough engineers to avoid considerable losses of carbon dioxide by leakage.

Fig. 14 shows a 15-atmosphere mercury pressure gage. It has five mercury columns, each having a capacity of 3 atmospheres pressure; it can thus be used to measure pressures up to 15 atmospheres. Mr. Meyers has spent much time in getting that gage in working order.

Back of the gage is the balance used for the weighing of

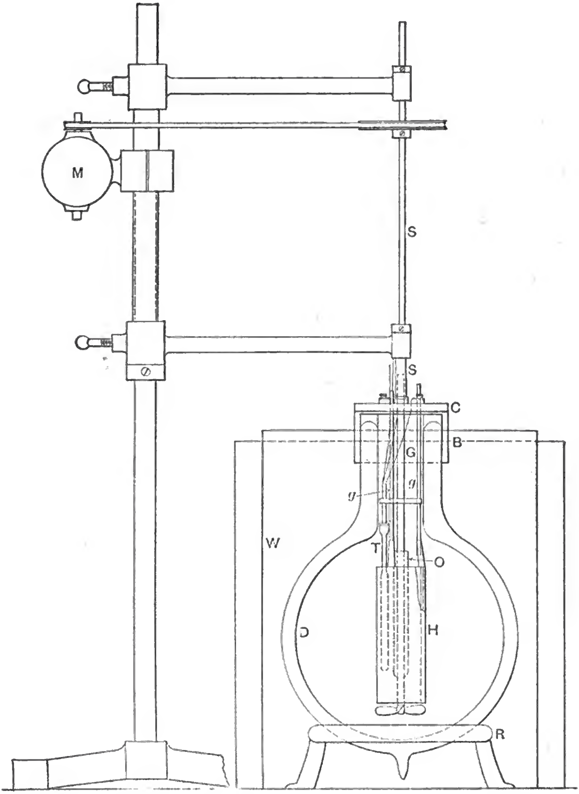


Fig. 12.—Calorimeter for Measurement of Specific Heats of Brines

ammonia bulbs as described by Dr. Harper, and for other problems requiring accurate weighing.

Fig. 15 is a 100-atmosphere piston pressure gage designed at the Bureau. It has been completed but has as yet not been used, since it has not been needed for any of the problems undertaken so far. It will be in demand when we come to study the constants of carbon dioxide. I might call attention to the fact that in the design of this gage the same principle was employed as Mr. Fairbanks

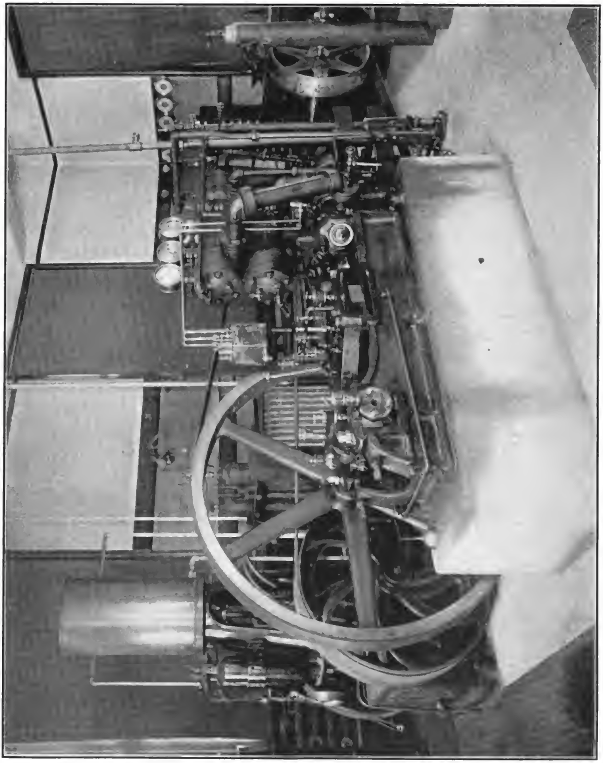


Fig. 13.—The special CO_2 Compressor Furnishing the Cooling for the Experiments

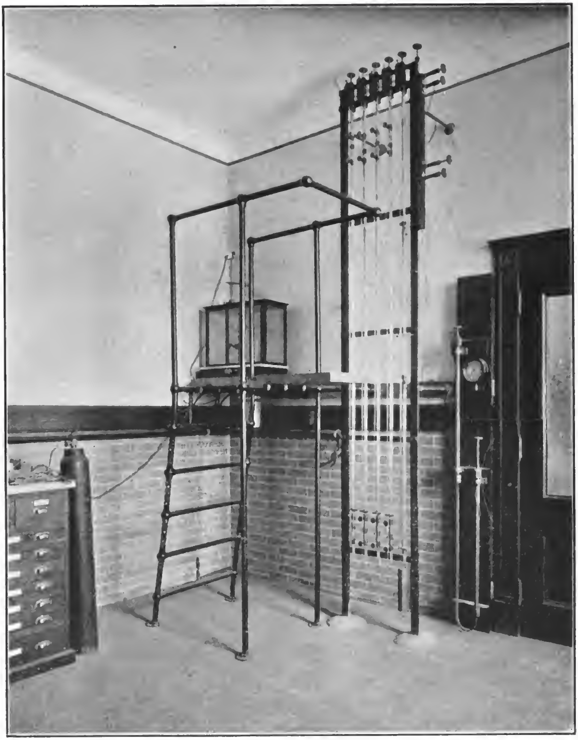


Fig. 14.—Primary Standard Mercury Pressure Gage. For Vapor Pressure Measurements Requiring the Highest Accuracy and for Calibrating Secondary Gages. Range 0 to 225 Pounds per Square Inch.

described in the building of his safety valve; that is, the weight is applied at a point below the bearing of the piston in the cylinder, although the weights are supported at some distance above this point. The whole piston is an extremely good fit in a hardened steel cylinder. There is a ball in the bottom of the hollow piston supporting a plunger which has another ball joint above, so that weights are applied on a support hung freely on the upper ball joint, and the thrust is transmitted to the piston through the plunger and the lower ball joint, leaving the piston bearing entirely free from side pressure.

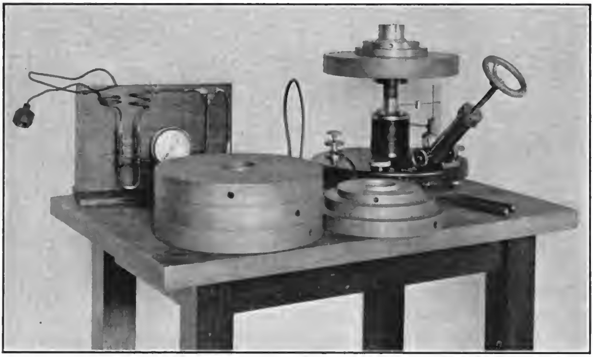


Fig. 15.—Piston Pressure Gage. For Accurate Measurement of Pressures up to 1,500 Pounds per Square Inch

The piston has an area of one square centimeter, *i. e.*, a little over three-eighths inch in diameter, and the total weight used amounts to about 200 pounds.

C. H. Herter—Do you use liquid air in the Bureau of Standards?

Mr. Dickinson—Oh, yes; we occasionally use liquid air in much of this work. Every few days there is a demand for liquid air, for some purpose or other, so we produce it at the Bureau of Standards in our own plant.

I believe that this includes the main problems now in progress under the appropriation for Constants of Refrigeration.

President Doelling—Gentlemen, the papers read to-night have given us a glimpse of the enormous amount of scientific research work done by the Bureau of Standards, and we refrigerating engineers, not only of this country but of the whole world, are deeply indebted to Dr. Stratton and his staff for the effective and valuable work they are doing, and I think we should show our thankful recognition for this work by extending a vote of thanks to Dr. Stratton and his staff. A motion extending a vote of thanks was adopted.

The Secretary—I think we might have a word from Dr. Stratton, he is here and is a pretty good talker.

S. W. Stratton—I always hesitate to say anything about the Bureau in the presence of any of the members of my staff, because they know so much more about it than I do. I could not help thinking this evening of how this piece of work illustrates a number of things that you know very well, but perhaps have not thought of it in just that light.

First, there is the cooperation between scientific men in working out a problem of this kind. One of the reasons why the Bureau has succeeded in many of these problems is that there is a spirit of cooperation there; they help each other; a man does not go off in a dark room and come out in a few years with his solution of a problem; he goes to the electrical expert for advice when he somes in contact with that kind of work or to the heat expert when he needs advice on that subject. If these gentlemen had been compelled to do all of the work themselves, to build up all the standards used, it would have taken a very much longer time to do this work. That sort of cooperation is going on at the Bureau to-day to a greater extent than in any other scientific institution, and I think I may say, without undue boasting, that that is the point in which the Bureau of Standards excels the foreign institutions of the same kind, there is greater cooperation between the scientific workers.

Take, for example, these experiments; where would they have been without the aid of the chemist? We have a very close working relation at the Bureau between the chemists and the physicists you do not find in any other institution I know of; it is especially not the case in our university laboratories; the chemist as a rule thinks it is an imposition to call upon him for pure mattrials to use in the physical laboratory. I have been through this, I know what it is, and in the Bureau of Standards the reverse is true.

This also illustrates the great value of cooperation with the industries in the solution of scientific problems. I have often quoted this piece of work as being an ideal example of how the scientific laboratory ought to work in connection with the industries. I know from my contact with these people, the men who are doing this work, that the greatest incentive they have had to produce good results, is your interest in it and your appreciation of these results. We never take up a piece of work of this kind at the Bureau without gathering about us representative men, usually in just this way, *i. e.*, through the technical societies or associations of that particular field of effort. There are perhaps 30 or 40 such associations working with the Bureau to-day—too many of them to mention.

This work illustrates another point which few practical men think of; that is, the necessity for scientific work of the very highest order in the industries. The man who says that an approximate determination of a constant of this kind is good enough for practical purposes does not know what he is talking about; the determination that is needed is the most accurate possible; then if you only need two decimal places, throw the rest away; but you want to do the throwing away, you want to know what the real value is, so far as it is possible to obtain it, and that is true throughout this kind of scientific work, it must be done with all the accuracy possible. I predict that it will be a long time in the future before these results are surpassed in accuracy, and yet they are none too accurate for your purpose. Much of the distinction between practical and scientific work is nonsense, there is little distinction, both often require the utmost accuracy. In both cases an approximate result is sometimes sufficient.

This also illustrates another thing I desire to call to your attention. I felt rather sorry for Dr. Dickinson as he talked to you in common units of weights and measures, as his measurements were all made in the metric system. Now that is a ridiculous state of affairs; why can't we all come to one common system of weights and measures? Out of courtesy to you gentlemen—probably it was not necessary, because I think most of you are as familiar with the metric system as we are—they were transformed into the common units; but that is always the spirit of the scientific men who come in contact with the industries, to give you the results in the units you are accustomed to use. If the engineer wants his results in pounds per square foot, we will give them to him that way, but how much better it would be if we would all come to a common system of weights and measures? I thank you very much for your appreciation of this work.

REVIEW

TRANSPORTATION OF REFRIGERATED PRODUCE

The Transportation of Refrigerated Produce. A. R. T. Woods, London, England. A paper read before the City of London College, March. 1915. 46 pp., 14 Figs.

In no department of refrigeration has such progress been made as in the over-sea transportation of perishable foodstuffs. Shipowners have provided highly specialized vessels for that purpose, and have equipped them with machinery which is a triumph of invention and engineering skill. Further, these delicate and costly machines are superintended by scientific craftsmen who require a special training for their work. How successfully the ship and her refrigeration and the engineers in charge co-operate is shown by the fact that so large a part of the nation's perishable food supplies is regularly brought to these shores in good condition, after passing through the vicissitudes of temperature encountered on the long ocean voyage from our New Zealand and Australasian possessions or the River Plate, etc.

But the same care and attention which renders these voyages possible are to a great extent conspicuous by their absence at the hands of those responsible for the passing on of the perishable foodstuff from the wharf or ship side to the consumer. Why this should be so it is difficult to understand, except on the ground that the British nation in its onward march in the direction of utilitarianism is handicapped by that spirit of insular conservatism which for good and for bad is part of the national character.

The carriage of perishable foodstuffs by sea is a science of quite recent development, as the following summary of the milestones in this important industry will show:

First consignment of chilled beef from U. S. A. to England (by ice and salt process)	1874
First consignment of frozen meat from New Zealand to England (by mechanical cold air process) per steamship *Circassian*	1879
First consignments of frozen meat from Australia to England (by mechanical cold air process) per steamship *Strathleven*	1879
One of the first consignments of chilled beef from the Argentine to England (by mechanical brine system), per steamship *Highland Lassie*	1893

The first adaptation of the ice and salt principle for the carriage of chilled beef from U. S. A. to this country, comprised a plant consisting of a large wooden tank, lead lined, placed in a convenient position adjoining the insulated 'tween deck in which the beef was hung. This tank was filled with broken ice, sprinkled with coarse salt.

The drainage or brine from the melting ice was drawn off from the bottom of the tank by a small pump, and circulated through iron pipe coils extending over the roof of the meat chamber. After circulating the brine round the chamber, any surplus was discharged into the ship's bilges.

The carriage of meat by this process was very satisfactory, but the great difficulty to contend with was the great consumption of ice, especially when the ship encountered the higher temperatures obtaining in the gulf stream.

The most popular system of which Messrs. Swift were the chief exponments was very popular with American shippers, and it was only abandoned on the introduction of mechanical refrigeration.

Figure 1 shows the general arrangement of a meat chamber on the ice and salt system fitted on shipboard.

I will now pass on to the modern refrigeration plant on shipboard, and will briefly outline the principal methods applied, viz.: "The Cold Air Battery System" and "The Cold Brine System."

Figure 2 shows the general arrangement of the cold air system and the disposition of air ducts, etc., which will enable you to follow the flow of

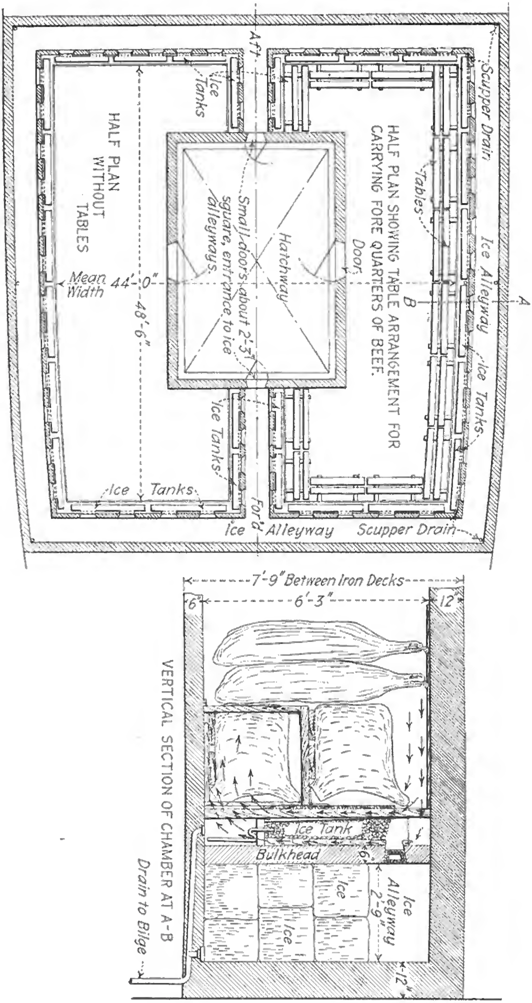
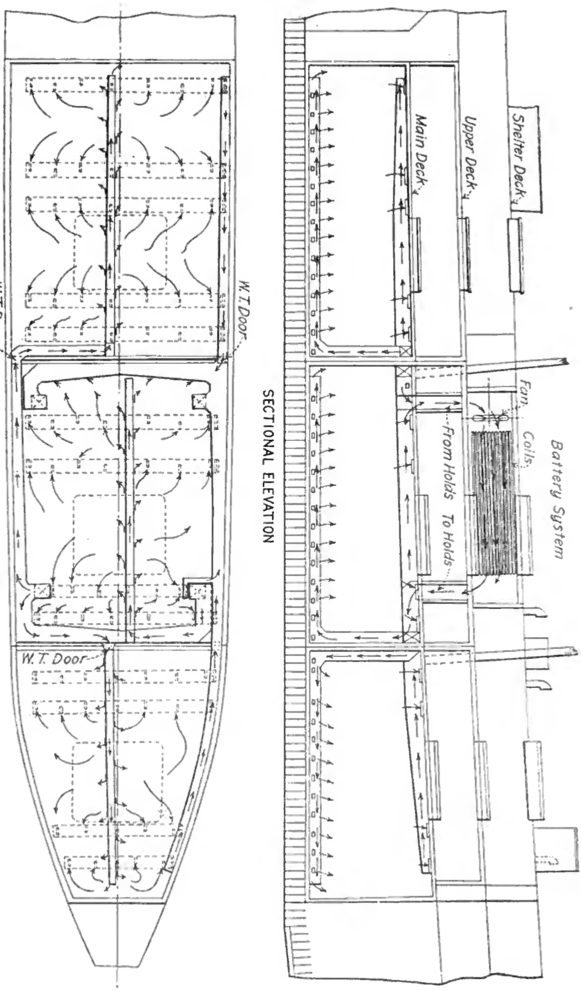

HALF PLAN WITHOUT TABLES
HALF PLAN SHOWING TABLE ARRANGEMENT FOR CARRYING FORE QUARTERS OF BEEF.
Aft
For'd
Ice Tanks
Ice Tanks
Ice Tanks
Ice Tanks
Tables
Scupper Drain
Scupper Drain
Ice Alleyway
Ice Alleyway
Hatchway
Small doors about 2'-3" square, entrance to ice alleyways.
Door
Mean Width 44'-0"
48'-6"
A
B
7'-9" Between Iron Decks
6'-3"
6"
12"
Ice Tank
Bulkhead
6"
Ice
Ice
Ice Alleyway 2'-9"
12"
VERTICAL SECTION OF CHAMBER AT A-B
Drain to Bilge

Fig. 1



SECTIONAL PLAN

SECTIONAL ELEVATION

Fig. 2

cooled air from the battery coils, a detailed arrangement of which is shown in Figure 3, to the various ship's chambers, and returning again to the battery, thus completing the cycle. The refrigerating machinery is erected preferably on the shelter deck; on the upper decks are the air coolers, one on each side of the ship, containing the fans for circulating the air and the coil batteries through which ammonia is expanded at a low pressure.

I will now pass on to the brine pipe system as now adopted on the modern meat-carrying steamers.

With this system, instead of air ducts being distributed about the chamber, there is secured to, and extending over the surface of the roof, sides and ends of the chambers, brine pipe grids through which cold brine is circulated, thus supplying the cooling medium, the air circulation being a natural one. To maintain a uniform temperature the chambers must be well piped; that is, in my opinion, you can scarcely have too much cooling surface. A chamber well piped has always a command of the articles stored, and allows a larger quantity of brine to be passed through the circuits at a lower velocity. A most important point in the piping arrangements of a chamber is the sub-division of its circuits. These circuits should be controlled from an inlet and outlet header, governed by a flow meter, an inexpensive instrument, which gives the number of gallons of brine flowing through each circuit, and indicates any trouble that may arise in the circuit.

On the question of brine circuits there are two arrangements now used, viz.: the open circuit system and the closed circuit system.

OPEN CIRCUIT SYSTEM

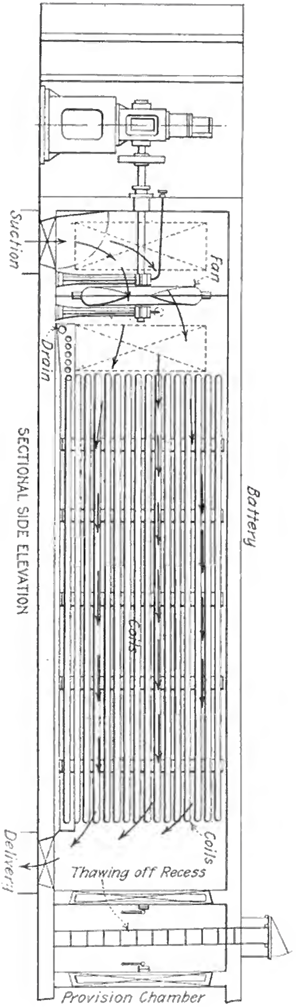
In the open system, which is in use on some steamers carrying refrigerated cargo, the brine, after passing through the chamber, returns into an open tank, which is divided into three compartments to receive the freezing, chilling and thawing brines, respectively. When at work, for example, on a chamber containing frozen goods, the one-way open bottom cocks which are fitted between delivery headers are open to the frozen header and the chutes placed as shown on cross section, sloping into frozen tank, Figure 4, so that with the frozen pump in service, having its suction connected to tank bottom, the brine is pumped through refrigerator, where it is cooled, from there passing through mains into header, next down into the chamber, where the refrigeration takes place, and finally finds its way through returns, passing down the chutes into frozen tank, where it is again pumped into evaporator and cooled, thus completing the cycle.

In the thawing arrangement an independent heater is so fitted that when at work the chilling inlet valve on the brine main is closed, and the thawing valve open, as shown in diagram. The hot brine is now pumped through chamber in the usual manner, finally returning down the chute into the thaw tank, where it is drawn off by the pump and again discharged through the heater.

The next diagram, Figure 5, shows very clearly the general arrangement of the closed circuit system of brine distribution, which perhaps is the most modern arrangement.

CLOSED CIRCUIT SYSTEM

The advantages over the open system being, viz.: the dispensing with brine tanks entirely, thereby saving over 50 per cent of space required for the brine distributing room, and, further, after the system is once fully charged with brine, there is little chance of air being taken into the system while in operation, or when the plant is idle. With this arrangement there are also three brine mains, each supplying brine at different temperatures required for distribution to the various chambers, which may be carrying either frozen or chilled cargo, or for thawing down the chambers when discharged. These mains are tapped by all the headers supplying the brine to the different chambers. If frozen brine is required for any particular chamber, the attemperator cock shown on the diagram (which is of special design, being an open-bottom cock with frozen and chilled orifices in the plug) is switched onto frozen



SECTIONAL SIDE ELEVATION

Fig. 3

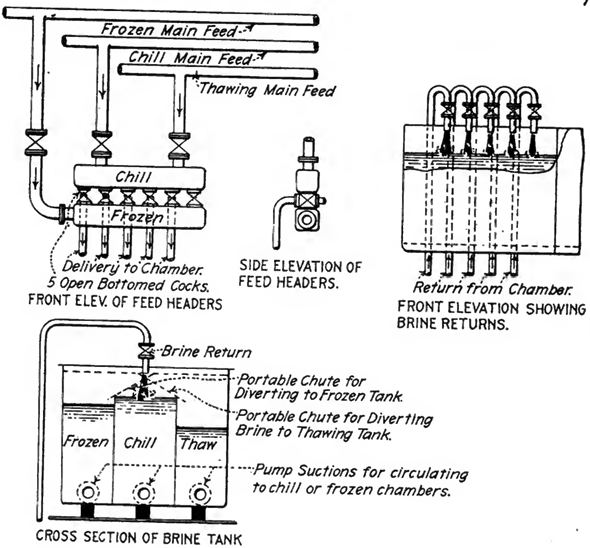

Frozen Main Feed
Chill Main Feed
Thawing Main Feed
Chill
Frozen
Delivery to Chamber.
5 Open Bottomed Cocks.
FRONT ELEV. OF FEED HEADERS
SIDE ELEVATION OF FEED HEADERS.
Return from Chamber.
FRONT ELEVATION SHOWING BRINE RETURNS.
Brine Return
Portable Chute for Diverting to Frozen Tank.
Portable Chute for Diverting Brine to Thawing Tank.
Frozen
Chill
Thaw
Pump Suctions for circulating to chill or frozen chambers.
CROSS SECTION OF BRINE TANK

Fig. 4

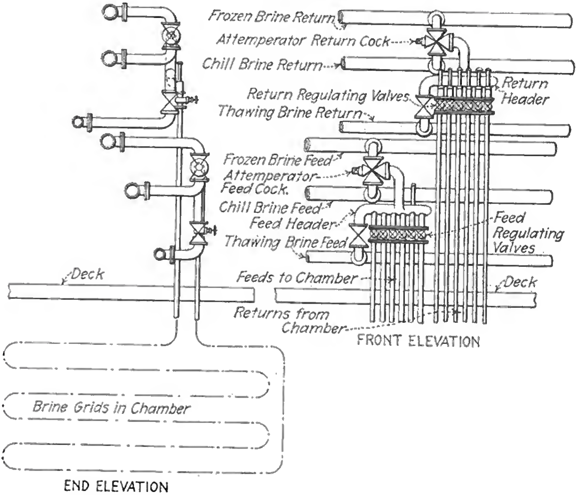

Frozen Brine Return
Attemperator Return Cock
Chill Brine Return
Return Regulating Valves
Return Header
Thawing Brine Return
Frozen Brine Feed
Attemperator Feed Cock.
Chill Brine Feed
Feed Header
Thawing Brine Feed
Feed Regulating Valves
Deck
Feeds to Chamber
Deck
Returns from Chamber
FRONT ELEVATION
Brine Grids in Chamber
END ELEVATION

Fig. 5

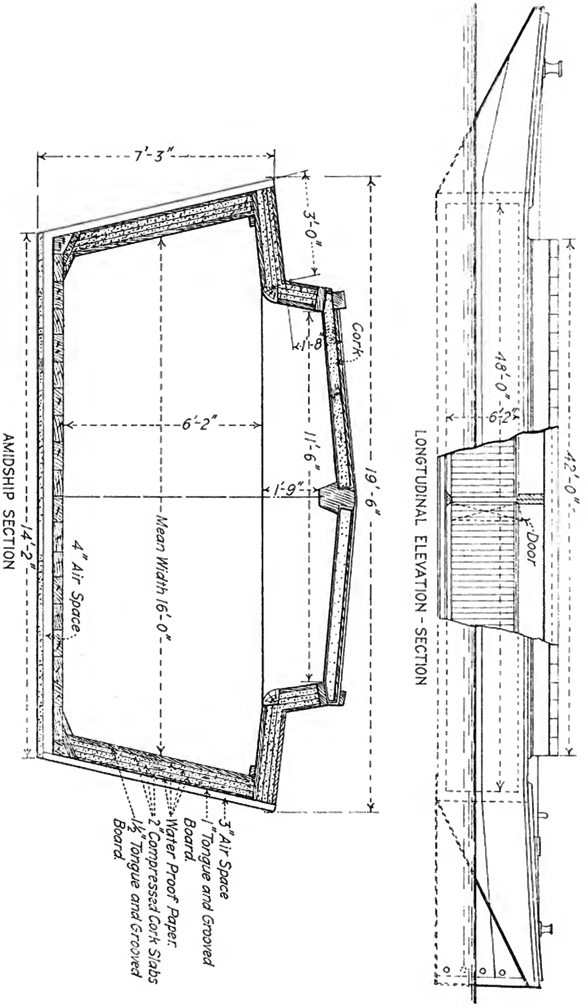

7'-3"
3'-0"
1'-8"
Cork
6'-2"
11'-6"
19'-6"
1'-9"
Mean Width 16'-0"
4" Air Space
14'-2"
AMIDSHIP SECTION
3" Air Space
1" Tongue and Grooved Board.
Water Proof Paper.
2" Compressed Cork Slabs
1½" Tongue and Grooved Board.
LONGTUDINAL ELEVATION - SECTION
48'-0"
6'-2"
42'-0"
Door

Fig. 6

main, thus taking the brine direct to feed header supplying that particular chamber. After passing through the different sections in the chamber it returns to return header, and hence through another attemperator cock, which is opened to the return frozen main. If chilled brine is required then the attemperator cocks are adjusted to chill mains, the brine then making a similar circuit as described in the frozen operation.

Should it be found necessary that any particular chill chamber should require a slightly lower temperature of brine than that supplied by the main chill feed, this can readily be done, first, by slightly increasing the pressure in the frozen main and then by setting the plug of the attemperator cock, so as to inject a little frozen brine with the chill brine supplying that particular chamber (the orifices of the plug being so arranged so that this can be done), the brine then going through the feed header is mixed, and at a lower temperature, hence through the sections in the chamber.

The surplus of brine caused by injecting the frozen brine to the chill brine is got rid of after passing the return header through the attemperator cock on the return mains, which is adjusted so as the surplus is returned into the frozen return main.

The writer proceeds to describe and show illustrations of past and present types of insulated delivery trucks, English and American refrigerator cars, ice and salt brine circulation, and ammonia direct-expansion systems for cooling them. He also comments as follows on the present-day insulated barge, such is is in use on the River Thames:

INSULATED BARGE

Figure 6 shows one of the best class insulated barges in use to-day, and I may say is far short in efficiency of what is required in these days for this class of transport, the main defects of this barge being, first, the excessive size of the hatch opening, which practically forms the entire roof of the insulated chamber, and which, being all in portable sections or hatches, is liable to allow a fair amount of leakage of warm air into the hold. Secondly, the floor of the hold is really not insulated at all, thus allowing a great transmission of heat from the water.

Let us just try and follow what happens to a cargo that is entrusted to the care of one of these craft. It arrives alongside the steamer, in what condition? It is very questionable if it could be termed, in the first place, clean, or having been cleaned, disinfected or washed after discharging its last cargo; if not the cargo (assume a valuable meat cargo) goes into a very questionable receptacle to start with. Secondly, if the weather should be warm and sultry, and the barge, perhaps, having been days out of use for a refrigerated cargo, it arrives, one might say, as a thermo chamber guaranteed to heat up its contents. Now what happens to this cargo? Immediately it is put on board it commences to absorb the heat out of the barge's insulation, and as soon as this transfer commences, it is only the question of the cargo's resistance until it comes to the point of temperature of its surroundings. If this cargo has luck, it might be delivered within 24 hours at the cold stores, or it may miss a tide, or have to wait its turn to discharge; in either of these cases it means at least 36 hours before delivery could be given.

It is not at all an unknown thing that consignments have been discharged in a very soft condition, if not worse.

All this, to my mind, is a crying shame to ordinary intelligence that such a state of things should be allowed to continue. The losses in the past must have been untold, and not only to the farmer or factory who first provides the goods, but also the shippers and consumer, not to mention the fact that all this loss could have been avoided, and undoubtedly help to affect the market price. One might say that the time has come when more notice should be given to these details by the authorities, and insist upon suitable craft being provided (and frequently official inspection of same) for the safe carriage of all such perishable articles.

Therefore, there is no doubt that pre-cooling and mechanical refrigeration requires introducing into all lighter and barge craft engaged in the transit of perishable commodities; then the difficulty of having the barge at a proper temperature to receive its cargo, and to maintain same, would be entirely overcome, and much waste of good food avoided.

The author closes with a strong appeal for means of eliminating the "economic crime" resulting from lack of adequate distributing facilities, and pays a pleasing compliment to the United States in saying that they "might take a leaf out of the book of the U. S. A., who have this important matter well in hand."

SOCIETY AFFAIRS

AUTOMATIC WATER SPRINKLER SYSTEMS IN REFRIGERATED ROOMS

The Society was represented at the meeting on March 7 of the Automatic Sprinkler Committee of the National Fire Protection Association by Gilbert Crawford, Jr., representing Louis Doelling, Chairman of our Committee on Municipal and State Regulations for Refrigerating Plants and Refrigerants. The following extract from the minutes of this meeting will interest many members of the Society:

"Dry Pipe Sprinkler Systems in Refrigeration Rooms—Ice formations of such a character as to seriously impair the automatic sprinkler protection have been reported in a number of cold storage warehouses, following detailed examinations of the piping by several inspection departments.

"Your Committee was requested to consider this subject, recommend a uniform method for conducting further tests, and solicit cooperation of the various inspection departments, with a view to ascertaining if these formations may be expected generally in risks of the class, and if so, if adequate preventative measures have been or can be devised. A uniform blank has therefore been distributed by the Committee among such of the active members as it should interest, and the promise of cooperation has been very satisfactory. As, however, the blank was not issued until after the first of the year, there has not been sufficient time to furnish the required information for any extended Committee report at this time.

"Technically, there is danger of such ice formation from the presence alone of the priming water of the dry pipe valve, ignoring entirely any moisture that may enter the system through the air compressor or by other means. Until there is opportunity, however, to secure details from the further examinations, and to inquire particularly into remedial measures which have already been adopted and said to be effective, or that have been proposed, your Committee would prefer not to recommend any action through the Sprinkler Regulations. Permission is, therefore, requested to continue the matter further, and the cooperation of anyone interested in this particular subject is earnestly solicited. A copy of the uniform blank referred to is appended as a part of this report.

"Among the various measures that have already been adopted or recommended are: Calcium chloride reservoir in the discharge line from the air compressor, ammonia-cooled desiccator or freezer, oil seal on priming water of dry pipe valve, calcium chloride reservoir in the main riser above the dry pipe valve, separator in the discharge line from the air compressor, air suction from freezing room, by-passes in the sprinkler lines in which the ice crystals may build up, leaving the sprinkler pipes free, greater pitch to the sprinkler piping and use of a greater number of flanged connections, maintaining at least 1 foot air space where possible between the sprinkler piping and the refrigeration coils, or insulation of piping where in contact with or in close proximity to such coils, rearranging the present methods of air connections, connection of anti-columning pipe where used to be within the warm dry pipe valve enclosure, where feasible sprinkler piping for freezer sections to be kept entirely separate from balance of risk.

"In this connection your Committee would point out that any pre-drying arrangement is ineffective if the air connection enters the system in such a manner that the air will pass through the priming water, or any drainage that may lie on top of the priming water, as is extremely likely to be the case in the method adopted for air connection with some types of dry pipe valves. Particular attention is called to Rule 54 of the Sprinkler Regulations in this respect, which states that the air connection should enter the system in the main riser above the dry pipe valve."

If any members of the Society have any suggestions to offer along this line they are respectfully requested to communicate with Mr. Doelling.

TRADE CATALOGS

(Below is given a list of trade catalogs, etc., received at the Society's office since the last list was published in the A. S. R. E. JOURNAL)

Gifford-Wood Co., Hudson, N. Y.—Ice Conveyors and Tools.

Armstrong Cork & Insulation Co., Pittsburg, Pa.—Nonpareil Corkboard Insulation.

De La Vergne Machine Co., New York, N. Y.—

- Initial and Operating Cost of Large Ice Plants.
- De La Vergne Countercurrent Ammonia Condenser.
- De La Vergne Ammonia Purifier.
- Liquid Base Oil.
- Safe Ice Cans.
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PERSONAL

WALDEMAR R. KREMER was recently appointed General Sales Manager of the Vilter Manufacturing Co., Milwaukee, Wis. Mr. Kremer has been connected with the Vilter Company for nearly ten years as consulting electrical and mechanical engineer in the sales department. In his new capacity he will have general charge of sales and supervision of branch offices and agencies in this and foreign countries.

A. S. R. E. Journal

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PUBLISHED BI-MONTHLY BY

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

154 NASSAU STREET, NEW YORK, N. Y.

Edited by the Secretary, under the direction of the Publication Committee.

Reprints from this publication, which is copyrighted, may be made on condition that the full title of paper, name of author and date of presentation to the Society are given.

Subscription price, United States and dependencies, $3.00 a year; Foreign countries in Postal Union, $4.00 a year.

Entered as second-class matter, February 11, 1915, at the Post Office at New York, N. Y., under the Act of August 24, 1912.

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A. S. R. E. Journal

Vol. 2 May, 1916 No. 6

A THEORY OF COOLING TOWERS COMPARED WITH RESULTS IN PRACTICE*

B. H. Coffey and George A. Horne, New York, N. Y.

THE 1915 OPERATIONS

As outlined in our paper[1] last year, our observations upon the cooling tower were continued throughout this season.

As a result of last year's experience we arrived at the conclusion that certain apparent inconsistencies in the results could be smoothed out by taking, if possible, all readings at the site. In consequence of this opinion we made arrangements early in the season with the local United States Weather Bureau Station to install a standard anemometer upon the tower, which was done, and has given a continuous record of wind velocity.

It was also our intent to install a recording wet and dry bulb thermometer, but, unfortunately, we were not able to accomplish this before October 6, since when our record of all measurements taken at the site has been complete. Prior to October 6 we used the United States twenty-four-hour record from the Arsenal Station, Central Park, New York, N. Y., of wet bulb temperatures derived from a hair hygrometer, occasionally checked by a sling psychrometer, and believed to be fairly accurate. The use of this record, we believe, at times introduces discrepancies of considerable magnitude, due to differences in wet bulb temperatures at the two localities, and though the general averages agree fairly well, specific observations at identical hours may differ widely, thus introducing apparent inconsistencies in the record that have no existence in fact, and we therefore decided to discard this part of the data and use only the readings subsequent to October 6, when all quantities were measured at the site. In following this course, we, unfortunately, lost the benefit of the hot-weather readings, which would have made the comparison with theory much more complete than we are able to present.

Fig. 1 is a view of the tower, showing the anemometer at one end and the shelter for the recording wet and dry bulb thermometer at the other end. We also thought a general view of the tower might be of some value in connection with the subject matter of the paper.

---

* Read at Eleventh Annual Meeting, New York, N. Y., December 6, 7 and 8, 1915.

[1] A Theory of Cooling Towers Compared with Results in Practice. B. H. Coffey and George A. Horne, New York, N. Y. A. S. R. E. Journal, November, 1914, and January, 1915.

The temperatures of the incoming and outgoing water were taken by Taylor recording instruments throughout the season, and the quantity circulated by a Venturi meter, as before.

We wish here to call attention to our wet bulb readings, probably the most important temperature affecting cooling-tower results. The true wet bulb temperature can only be obtained when there is sufficient relative movement between the bulb and the air to thoroughly sweep away the aqueous vapor as it is discharged from the evaporating surface. With the stationary or Mason instrument

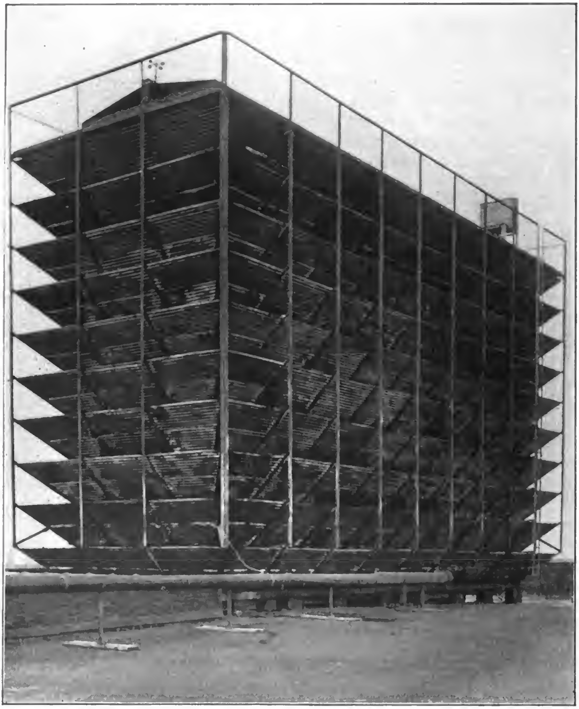

Fig. 1

this is fully accomplished when it is exposed to a fair breeze, but on still days, or if the instrument is located in an enclosed position or on the lee side of some obstruction, as a wall or building, the readings will be much too high, and any conclusions based thereon are erroneous. We call attention to this, as this instrument is very popular, both on account of its moderate price and the apparent ease with which the wet bulb temperature may be obtained. If a stationary wet bulb is used it should be in the draught of an electric fan or fanned by hand most vigorously for several minutes before taking the reading. Our recording wet bulb instrument referred to was in the constant draught of an electric fan.

For general use on cooling-tower work we recommend the whirling wet bulb or sling psychrometer of the United States Government pattern, and that the directions for its use as given in the Government Bulletin be strictly followed. With particular reference to cooling-tower work, would supplement these suggestions by recommending that all wet bulb readings should be taken to windward of the tower or at some point where the large volumes ot aqueous vapor discharged from the tower do not affect the quality of the atmosphere. and thus vitiate results. If some standard form of procedure for taking wet bulb readings is not adopted, results affected by them cannot be compared, and much engineering data that should be of great value to the profession may become misleading and of positive injury.

THEORY

Further consideration of the theory of atmospheric cooling propounded in our paper last year has led to the following conclusions:

Considering the difference between the final water temperature and the wet bulb temperature, or approach to the wet bulb, as a measure of tower efficiency, the degree of approach is affected by four variables—

1. The cooling range, or number of degrees of heat extracted in the tower.
2. The position of the wet bulb in the thermometric scale—a wet bulb temperature that is high on the thermometric scale can be more closely approached than one at a lower temperature level.
3. The ratio by weight of air to water in contact throughout the cooling process.
4. The ratio of exposed water surface to the weight of water cooled in unit time.

Any consistent theory must therefore be capable of expressing the effect of these variables upon the terminal temperature, which, less the wet bulb temperature, is the wet bulb approach.

Plates 1, 2 and 3 are characteristic curves derived from our theory, which show the effect of these variables on the cooling water. On all three plates terminal temperatures are given at constant cooling surface 13.8, and on Plates 1 and 2 the effect of variable cooling surface is indicated by giving the temperatures existing at cooling surface 4, 6, 9. These plates are all drawn to the same heat scale

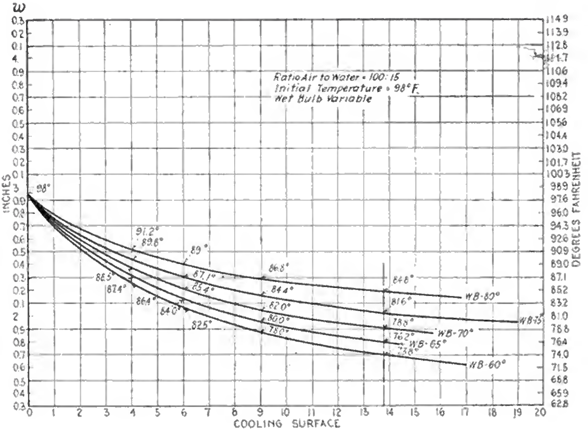

Plate 1

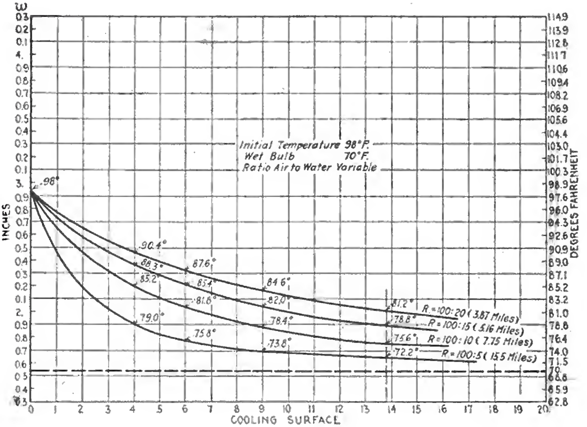

Plate 2

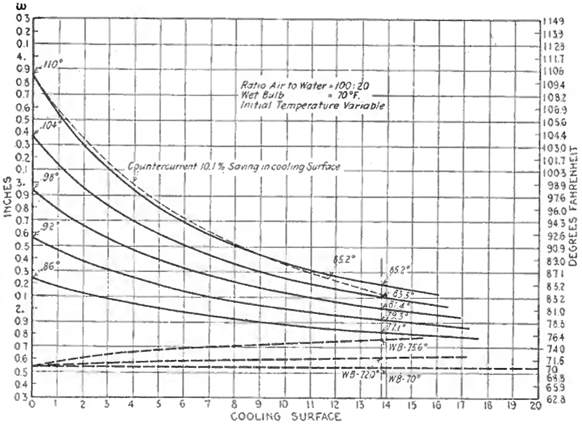

Plate 3

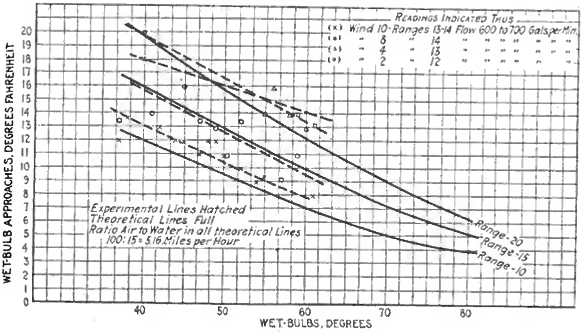

Plate 4

used with Plate 4 in our first paper, and are therefore comparable both with it and with each other. The curves show the temperature gradiant for parallel current heat transfer except the dotted curve of Plate 3, which shows the application of the theory to counter-current transfer.

The corresponding curves of the heating air have been omited except on Plate 3, upon which is shown the air curve corresponding to the highest and lowest water curve. The air curves are shown in hatched lines.

In all the plates the vapor temperatures corresponding to the values of the function ω have been entered on the right-hand side.

The steps taken in calculating the curves are somewhat tedious, and were explained in our first paper.[2]

We have not yet succeeded in developing general formulæ for determining these curves from the variables, which would enable us to solve for final results at once and so render the problem much less unwieldy to handle and practicable for general professional use, and our next task will be directed to this end. Such results when obtained will be communicated to the Society.

In order to compare the theory with practice it was necessary to prepare curves of theoretical results referred to wet bulb approach as a standard, in which the effect of the selected variable is shown through a wide range. Plate 4 shows such curves in which the wet bulb is the variable. Three curves are given for constant ranges 10, 15, 20, the air to water ratio 100:15, and the cooling surface 13.8 being constant throughout.

Plate 5 shows four such curves in which the air to water ratio is variable, and expressed in miles per hour, range, wet bulb and cooling surface being held constant.

These general curves were derived by calculating a large number of the cooling curves similar to those given on Plates 1 and 2, at different initial temperatures, thus giving the terminal temperatures and wet bulb approaches for various ranges. These results were in turn plotted against wet bulb approach and range and curves of constant wet bulb and air to water ratio obtained. From these curves in turn were taken out the wet bulb approach at the ranges desired for Plates 4 and 5.

The curves of Plate 4 show a gradual, almost straight line, increase in the approach to the wet bulb as the temperature level of the wet bulb rises until about wet bulb 70°, when the rate of increase lessens showing the tendency, in strict accord with the general theory, that with finite cooling range and rising wet bulb the approach will indefinitely increase, but never can coincide with the wet bulb itself.

Plate 5 shows the violent fluctuations in wet bulb approach caused by changes in wind velocity or air supply, but the curves show the same tendency to indefinitely approach the wet bulb, but never reach it, as exhibited in Plate 4.

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[2] A. S. R. E. Journal, November, 1914, p. 87.

COMPARISON OF THEORY AND PRACTICE

When a tower in commercial operation is used to demonstrate a theory, as in this case, none of the variables are under control, and the only alternative is to obtain an enormous number of continuous readings as conditions change, due to atmospheric fluctuations and the demands of the plant. The readings must then be all checked and segregated under the proper variables before final plotting, involving a large amount of clerical labor. It was due to this and our desire to extend the readings as late in the month as possible that we were unable to get our material in shape early enough for the November number of the A. S. R. E. JOURNAL, and thus give the members interested in the subject ample time to digest the paper for discussion. We trust, however, to receive written discussion, which will appear in subsequent issues of THE JOURNAL.

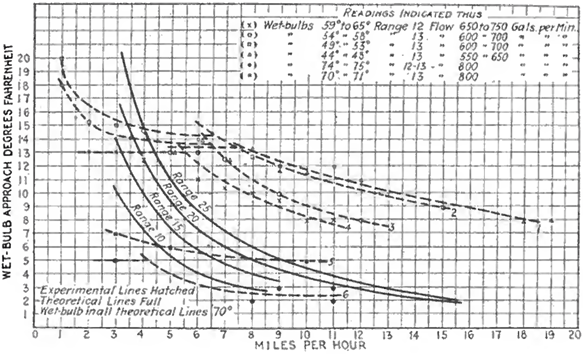

Plate 5

On Plate 4 we have plotted points under various winds and ranges as indicated on the plate. It will be noted that the medial lines of these points are essentially parallel with the theoretical curves, thus showing that the law of variation as expressed by theory appears to be substantiated to the extent of these experiments. The loss of our hot-weather record, already referred to, deprives us of the opportunity to extend points farther down the theoretical lines.

The experimental points on Plate 5 appear also to follow the characteristics of theory, except the sudden departure shown at about wind 6 and 7. This variation we attribute to the effect of natural draft, which begins to grow important when the winds become light and in effect, for a considerable period of light wind velocities, maintains the air supply practically constant.

Experimental lines 5 and 6 were taken from the discarded hot-weather record to indicate approximately the results at higher wet bulb.

You will note that, in spite of all the care we have taken in obtaining the experimental points, inconsistencies and discrepancies are still obviously present, due, we believe, to the impossibility of expressing with the recording instruments the exact conditions that exist. Still, in our opinion, the main trend of the effect produced by the variables is quite plain and, we submit, in general conformity with the theory we advance.

Further readings next season, with all measurements at the site, will enable us to accumulate valuable hot-weather data for further comparison with theory, but they can only more firmly establish its probable soundness; to completely prove it, will require experiments with a tower in which all of the four variables, referred to under theory, can be exactly measured. This is not possible with the atmospheric tower we have so far used in this investigation, which employs the drop and splash form of exposed water surface. In this type of tower the surface is made up of two elements, 1, fixed wetted surface; 2, water particles varying all the way from the standard drop determined by the viscosity, surface tension and temperature of the water to extremely small particles, into which the drops break up after impact with the decks. It is thus second element of surface that has defied our attempts to estimate. We therefore propose during the ensuing season to obtain the use if possible of a forced-draft tower employing wetted surface only, as it is sometimes termed, film surface. In such a tower the total wetted surface is, of course, easily obtainable, and, if facilities are provided for accurately measuring the air supply, the quantity of water circulated and for taking the necessary temperatures, all of the variables become known and the actual results in final water temperature can be directly compared with those calculated, thus establishing a criterion by which the theory can be demonstrated or destroyed.

The results of such trials, when made, will be communicated to the Society for further consideration.

THE CONSTANT THETA (θ).

In compliance with the statements in our paper of last year, we have made a number of direct determinations of this constant with the somewhat crude apparatus shown in Figs. 2 and 3, in which:

A — is the evaporator and its thermometer.
B — the wet bulb thermometer.
C — the dry bulb.
D — a check, combined wet and dry bulb thermometer or sling psychrometer.
F — an electric fan to ventilate the evaporator and wet bulbs.
G — a rectifier for changing the spiral air currents from the fan, to approximate rectilinear streams.
H — a fairly delicate scales reading to one-tenth gramme, and from which the evaporator is permanently suspended.

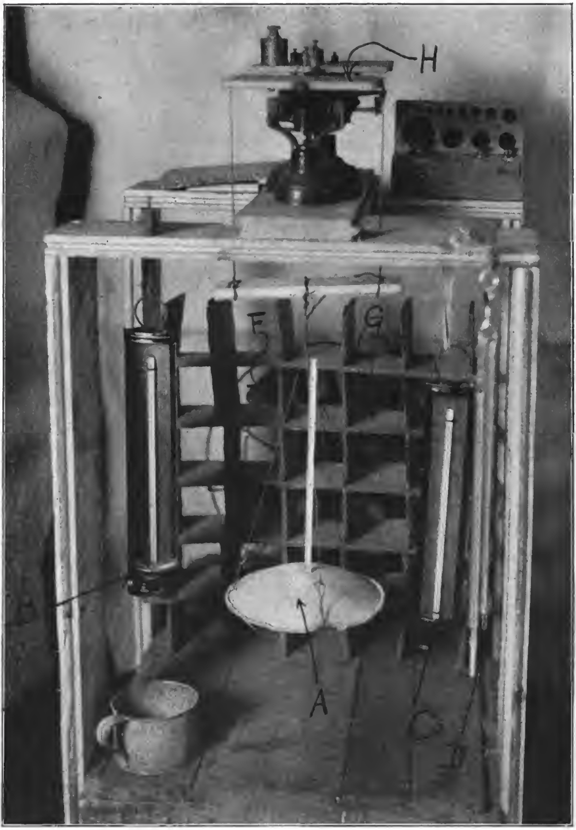

Fig. 2

A section of the evaporator is shown in Fig. 4. It is essentially two copper saucers, joined and covered with cotton cloth of considerable capillary power. Water in the lower saucer is fed by the circular wick to the edges of the cotton coverings, and from thence spreads over the entire surface, which remains completely damp until the water is exhausted.

The temperature of the interior of the evaporator will eventually fall by conduction to that of the two exterior surfaces, and this temperature is measured by the inserted thermometer. To repeat from last year's paper: The constant θ is the total heat loss per hour per square foot of surface per mercury inch difference of pressure between the values of the function ω at the cooling surface and at the wet bulb, and this difference in the values of ω is the measure of the

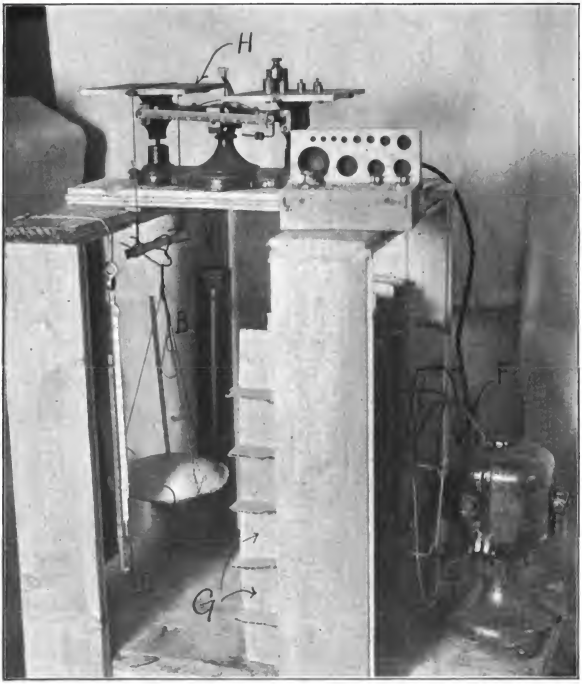

Fig. 3

total rate of heat loss from the cooling surface to the atmosphere. When the cooling surface itself sinks to the wet bulb temperature the difference in the two values of ω becomes zero, indicating one of two things, 1, either there is no further loss of heat to the atmosphere, or, 2, that there are two equal currents of heat in opposite directions; both of these states would satisfy the condition of no loss of heat. The second state is that which exists, as shown last year, the incoming current being sensible heat due to a temperature difference, the outgoing current latent heat of evaporation due to a pressure difference.

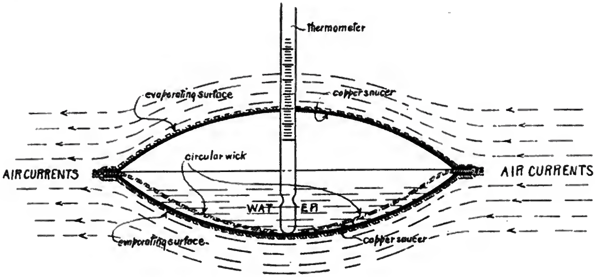

Fig. 4

To evaluate the constant θ it is necessary to determine the weight of water evaporated in known time from a known superficial area, under a known difference of vapor pressure expressed in mercury inches. These conditions are easily satisfied, provided the evaporating surface can be maintained at the wet bulb temperature. Under these circumstances the outgoing heat flow is entirely of the latent form, and it only becomes necessary to obtain the difference between the vapor pressures corresponding to the wet bulb and dew point temperatures, which difference is the pressure head causing the evaporation. This method of evaluating θ was adopted.

In order to obtain fairly accurate results, the superficial area from which evaporation is to take place must be of considerable extent, necessarily covering a body many times larger than the ordinary wet bulb, and to maintain such a body at the wet bulb temperature by its own evaporation presented something of a problem. Two theoretical conditions, which rigorously hold good in practice, had to be satisfied:

1. The entire superficial area of the body must be a uniformly active evaporating surface, for if any part of the area was not evaporating there would be an inflow of sensible heat at this point unbalanced by a corresponding outflow of latent heat, and the temperature of the entire body would rise above the wet bulb to a new point of equilibrium.

TABLE I.

SUMMARY OF EVAPORATION TESTS TO DETERMINE THE CONSTANT θ. θ=BTU. PER HOUR PER SQUARE FEET PER INCH MERCURY
TOTAL SURFACE IN EVAPORATOR, 83.8 SQ. IN.=0.58 SQ. FT. AVERAGE VALUE OF $\theta=652.5$ BTU.

No. of Test	1915.	t Dry Bulb °F.	t' Wet Bulb °F.	t'' Dew Point °F.	Evaporator °F.	Total Evaporation in Gram.	Total Time of Run in Minutes.	Relative Humidity %	θ—Btu.	Location of Test.
		1	2	3	4	5	6	7	8	9
1.....	Aug. 28	82.6	65.4	55.0	66.4	27.4	52	30	664	Engine Room
2.....	Sept. 6	91.2	76.6	70.6	76.8	21.0	45	50	679	"
3.....	" 7	90.2	74.1	66.8	74.7	21.9	45	47	636	"
4.....	Oct. 5	69.0	66.0	64.0	65.9	24.8	209	85	671	Open Air.
5.....	" 6	67.2	57.3	49.0	56.9	19.2	60	53	636	"
6.....	" 8	65.3	58.4	53.0	58.0	47.0	202	66	655	"
7.....	" 9	65.0	53.7	44.0	53.2	57.3	165	47	641	"
8.....	" 11	63.8	53.9	45.6	53.2	46.3	153	57	662	"
9.....	" 14	69.0	65.2	63.0	64.9	20.4	174	81	633	"
10.....	" 15	70.1	64.0	60.3	63.4	34.9	180	72	659	"
11.....	" 16	64.5	56.4	50.4	55.9	16.5	70	61	641	"

2. As considerable volumes of aqueous vapor are thrown off, which, if not promptly and fully removed, would raise the wet bulb temperature in immediate contact with the surface and so retard evaporation, it became necessary to adopt a form of surface that would be fully swept by the ventilating air currents and eliminate entirely the formation of eddies and dead spots, the effect of which is similar in principle to non-evaporating areas, as explained above. This condition was satisfied by adopting the lenticular shape formed by the two hemispherical cups—the direction of the enveloping air currents is shown at Fig. 4.

It is, of course, possible to determine θ at temperatures of the evaporator higher than the wet bulb, but in so doing there is no assurance that the surface is evaporating uniformly, and the possibility of a serious error is introduced. As has been explained, the very fact that the evaporator attains the true wet bulb, itself proves that the evaporation must have been uniform over the whole surface, and the error referred to thus automatically eliminated.

The method pursued in making the tests was as follows: Water was introduced into the evaporator, *A*, Figs. 2 and 3, through the thermometer orifice, the thermometer replaced and the total weight of the evaporator and all attachments approximately balanced on the scales. The wet bulb thermometer, *B*, was then moistened and the fan, *F*, started up. The thermometer, *B*, also in the air current from the fan, would generally attain the wet bulb temperature within a minute, but, owing to the much greater quantity of water and material in the evaporator, much more time was required. In some cases when the initial temperature of the water in the evaporator was much above the wet bulb temperature, half an hour was required. When the wet bulb thermometer and the evaporator thermometer reached practical equality the weight of the evaporator was accurately taken and the wet bulb, dry bulb and evaporator thermometers read, the time taken and all readings recorded. The readings were repeated at intervals during the run, the intervals being closer as atmospheric conditions showed greater fluctuation. The thermometer readings were then averaged, and from the average wet and dry bulb temperature the average dew point temperature was taken from the United States Psychrometric Tables, thus giving all the data required for calculating the constant.

Eleven experiments were made, as shown in Table I, giving the principal quantities and the value of θ as determined by each experiment. The mean value of θ from the eleven experiments is 652.5 B.t.u. and the maximum departure of any single experiment from this mean is 4 per cent, which, we think, justifies us in believing we have arrived at a fairly close approximation of the true value. The evaluation of the constant θ makes it possible to apply the theory in its present form to any tower employing film cooling surface exclusively, as with this form of surface the wetted area in contact with the atmosphere can be accurately calculated.

In closing, we wish to express our appreciation of the interest manifested in our work by James H. Scarr, Director of the United

States Weather Bureau, New York, N. Y., and his assistant, Charles D. Reed, to whom we are indebted for installing a Government anemometer upon the tower, and also to J. Ely, of the Taylor Instrument Company, Rochester, N. Y., for the loan of the recording wet and dry bulb thermometer, which has proved a most valuable instrument.

DISCUSSION

Mr. Horne—I wish to say that my colleague, Mr. Coffey, has had more to do with the theoretical aspect of the cooling tower problem than I We worked together and collaborated, but the larger part of my time was devoted to the experimental and practical part of the work.

We were rather surprised to find, at certain periods during the summer, as much difference as we did between the readings taken at Central Park and those taken at the local Weather Bureau on the roof of the Whitehall Building. At times they differed by as much as six degrees. The record at the Central Park station is made with a recording hair hygrometer, but in checking this with the sling psychrometer, the readings agree so well that it is probably correct to say that at times there was a difference of five or six degrees between the wet bulb temperature of these two local stations. This difference followed no special law; at times it was low and at other times high. Eddies, currents and different factors influence this matter.

G. T. Voorhees—Do the figures on the curves in Plate 1 (of the present, 1915, paper) represent the temperature of the water flowing down the tower?

Mr. Horne—The curves are theoretical. In Plate 1 the wet bulb is variable, while in Plate 2 it is constant at 70 degrees.

G. T. Voorhees—How is the ratio of air to water obtained?

Mr. Horne—The ratio of air to water is obtained by calculation. Mr. Coffey will explain how. It was referred to in last year's paper.

Mr. Coffey—We assume a vertical section through the tower, say one foot thick; we know the quantity of water flowing over the tower per square foot, per minute, hence the total quantity of water per minute dropping through the section. Then we assume that the air blowing horizontally through the section, at the existing wind velocity, goes through that amount of water.

Mr. Horne—I might add that we appreciate the fact that there is some retardation of the wind going through, which vitiates to some extent the calculation of the exact ratio, but to practically demonstrate the exact quantity of air going through would be almost an impossibility, requiring the use of anemometers all over the tower, which it would not be practicable to read.

G. T. Voorhees—Does not the amount of surface enter in determining those values as well as the amount of water?

Mr. Horne—The amount of surface is taken from the theoretical curves on Plates 1 to 3.

G. T. Voorhees—What does that term "13.8 cooling surface" mean?

Mr. Coffey—As explained in our paper, that is just a relative figure. We have not been able as yet to reduce it exactly to square feet. All these curves are based on evaporation in other towers. We happened to arrive at the value 13.8 while calculating the curves in our paper last year, 1914, where we had assumed a certain range in the tower In plotting the curve, the terminal temperature happened to strike 13.8, and we retained that ratio of cooling surface throughout all these plates to make them comparable.

G. T. Voorhees—Does that 13.8 mean square feet of cooling surface, gallons of capacity, or what?

Mr. Coffey—It does not mean 13.8 square feet, it simply means that 13.8 on our scale indicates the cooling surface necessary to obtain a temperature reduction from 110 to 85.2 deg. F

Mr. Horne—This we cannot express in terms of square feet on a tower of this type.

Mr. Coffey—Possibly it would be better to designate that surface by 10, or 1, and then the intermediate figures would be proportional to the total.

G. T. Voorhees—Does not the capacity of the tower depend on the cooling surface?

Mr. Horne—Yes, certainly; but in these theoretical lines the actual amount of surface involved is not known.

G. T. Voorhees—Do you take the capacity of the tower into consideration at all?

Mr. Horne—It certainly is, as will be shown later when all these points that you are bringing up will be made more clear. It seems impossible to arrive at the total active cooling surface on any type of atmospheric tower where the water descends in drops from deck to deck; for the reason that a given drop of water in falling breaks up and splashes into a hundred particles, and those particles are being cooled. The actual measurable cooling surface is not the real surface or only surface that has to do with the cooling effect, therefore it is an arbitrary figure.

Henry Torrance—In other words, the curves presented in Plate 3 show the relative cooling obtained with the same surface, assuming different initial temperatures of the water to be cooled.

Mr. Horne—Exactly.

G. T. Voorhees—But I cannot follow your reasoning, unless you can give us some idea of how we can apply that value 13.8. Does it mean cooling surface per pound of water, per gallon of water, or something else?

Mr. Coffey—It means that in any tower, regardless of size, 13.8 represents the total effective surface. To find the temperature reduction resulting with but one-half that amount of surface, starting with the same initial temperature, follow down the ordinate 6.9 (one-half of 13.8) and the curves will give you the terminal temperature obtained with half the total surface. As already stated by Mr. Horne, we have not been able to take up the question of specific surface on this tower because of the drops.

H. H. C. Lasker—Referring to Plate 5, what happens when those curves reach zero wind velocity?

Mr. Horne—Theoretically, when there is no wind we have no cooling effect whatever.

H. H. C. Lasker—But practically we still have the benefit of the chimney effect of the tower.

Mr. Horne—The chimney effect, or natural draft, really comes in while the wind velocity is still about two miles per hour, as shown in Plate 5.

President Doelling—I doubt that the rate of cooling at the different velocities proceeds as shown by the curves in Plate 5; I think the lines are not drawn right.

Mr. Horne—We purposely drew those lines without trying to smoothen out the curves. You see where the points lie. Of course, lines could have been drawn another way, possibly. With some of these points we did not have as many observations as we would like to have had in order to more fully substantiate our conclusions.

President Doelling—We are very much indebted to Messrs. Coffey and Horne for giving us the results of their numerous experiments. Anybody who has experimented with a cooling tower knows that it takes a long time to get results and secure a sufficient number of readings to be able to draw conclusions. I would like to suggest that, in presenting such papers, a drawing be always included giving the exact dimensions of the tower and apparatus used in the experiments. If that is done, the results of the experiments can be used later on and compared with results obtained by others.

Mr. Horne)—That can very readily be done. Mr. Reed, from the United States Weather Bureau in New York, will favor us with a few remarks and suggestions as to the use of the sling psychrometer.

Charles D. Reed—The sling psychrometer is used by the U. S. Weather Bureau in cases where the stationary instrument alone is not sufficient. It is portable and compact, as you see in Fig. 5, consisting of a wet bulb thermometer extending a little below the dry bulb thermometer mounted on the same back. By carrying a little vial of water for moistening the wet bulb, it is possible in any situation to quickly measure the humidity conditions of the air. Having wetted the cloth, getting readings is comparatively simple, it being only necessary to whirl the instrument at arms' length, at the same time slowly walking along, for otherwise the temperature of the observer's body will often raise the readings of the thermometers by half a degree or more. This difference is inappreciable when calculating dew point, humidity and vapor pressure, but becomes of considerable importance when calculating wet bulb temperatures.

I may point out a few of the peculiarities which affect the securing of correct wet bulb readings. For example, in the summer we find that on a hot, dry day, with temperatures above 90 degrees F., the amount of moisture contained in the thin muslin covering of the

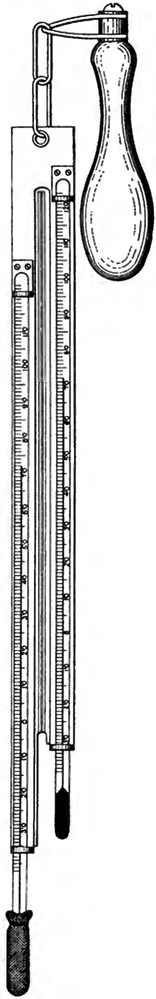

Fig. 5

wet bulb is sometimes not sufficient to reduce the temperature of the mercury in the bulb down to the correct wet bulb temperature; it then becomes necessary to supply means to insure reaching that point. An observer who is not aware of this condition will whirl his instrument until a constant reading is obtained, and take that to be the wet bulb temperature, when he has merely exhausted the supply of moisture on the bulb of his thermometer, which may be considerably above the true wet bulb temperature That can be obviated in different ways. One is, to use in the case of high temperature extra covering of cloth or wicking above the bulb of the thermometer, which will then receive more than the usual quantity of water, and sufficient to reduce the temperature of the mercury down to the true wet bulb temperature. Another method, often used where a watercooler is handy, is to start with water very near the wet bulb temperature. In that way I sometimes start with water below the wet bulb temperature, thus warming the mercury to the wet bulb rather than reducing its temperature to the wet bulb, the results being obtained in the same way by whirling until a constant reading is found. I think the latter method is the easier one, the other way being subject to some errors. For instance, if the same amount of extra covering and surplus of water was used with moderately low temperatures, inaccurate results might obtain by having too much water to handle. In other words, the evaporation of sufficient moisture to reduce a large mass of water and cloth covering to the true wet bulb temperature would be such a slow process that some observers might stop before the lowest possible temperature was obtained. For winter temperatures it is therefore best to have only a very thin covering on the wet bulb thermometer and a minimum amount of moisture.

When near the freezing point, moisten your thermometer with water, whirl it only a few seconds and take the readings quickly. The wet bulb temperature may drop to a very low point, as low as 20 degrees, without having congealed the moisture. It is often possible in this way to get the correct wet bulb temperature before the latent heat of freezing has been absorbed. If you do not stop whirling and get that reading before congealation takes place, it jumps instantly to 32 degrees and you have then the slow and laborious process of getting the evaporation from the ice surface, and that may require whirling for 5 minutes, before one is able to reach the wet bulb temperature again. To a layman it seems remarkable that water could ever be cooled to 20 degrees F. without freezing, but in our work we find that this is not unusual. In fact, instances are reported in the Journal of the Royal Meteorological Society of water existing at 12 degrees below zero F. without congealation. It is probable that the moisture we see in the air in the form of fog in zero weather is also uncongealed; it is still in the water stage.

These are some of the things encountered in taking readings, and by careful attention we can eliminate most of the errors. Owing to its convenience, portability and ease of manipulation, the sling psychrometer probably affords the most accurate means we have of getting wet bulb temperatures, while the figures obtained from a direct indicating hair hygrometer are subject to more or less error. In some instances, however, the hair hygrometer is highly accurate, and in the hands of an inexperienced observer, when used for temperatures below freezing, the results are probably more accurate than with the wet bulb, yet the hygrometer should be carefully checked by a man who knows how to adjust it properly and apply the corrections. We have at our Central Park branch office a hygrometer accurate within one or two per cent. over a period of several months. But, unexpectedly it develops an error, sometimes of five per cent., and has to be adjusted and corrected.

With reference to taking meteorological observations at the exact location where the cooling tower experiments are being conducted, I may say that I consider this of great importance. At the Whitehall Building our instruments are 414 feet above ground. There we sometimes notice below us a fog bank that is absolutely dense at the surface, while the sun is shining brightly up where we are. At such times there is surely a difference between roof and street of 10 or 20 per cent. in the relative humidity, and five degrees in the wet bulb temperature. You have seen the upper stories of high buildings, such as the Woolworth, enveloped in a bank of fog, while at the surface the atmosphere was comparatively clear. The differences in our readings at the Central Park station and the station at the Whitehall Building at the Battery are probably due to a combination of these conditions. For instance, take the wind velocities measured in November, 1915. At the Battery we have an anemometer mounted at Pier A, about 50 feet above the ground; we have another 454 feet above the ground on the Whitehall Building; and we have this other one at the cooling tower that Messrs. Coffey and Horne are testing. The one at Pier A showed only about 60 per cent. of the velocity obtained on the roof of the Whitehall Building, which is practically the same location, but 400 feet higher up, while the one at the cooling tower indicated only 51 per cent. Now, the elevation at the cooling tower is 162½ feet ostensibly, but as a matter of fact the surrounding buildings have raised the general level, so that the effective elevation above the surrounding level is probably less than half that amount. For the problem in hand it was therefore agreed that it was best to carry out the work on the spot where the tower is located.

The Weather Bureau is called upon to supply a vast amount of data to people engaged in cooling tower and humidifying work and other things affected by wind, temperature and humidity, but we always try to qualify our data with the statement that, while these are the conditions obtained at the point of observation, they may not strictly represent the conditions at the point where one wants to locate his heating, cooling, or other kind of a plant.

H. H. C. Lasker—We should appreciate the efforts of the authors of this paper to establish a theory of cooling towers. A theory, however, is not valuable unless it can be used as a basis for practical work.

The criticism which I would make of this paper is, that no theory has been proposed which could be used by refrigerating engineers to design cooling towers. This, in my estimation, is what is desired.

There are many variables which enter into the operation of a cooling tower, and in the paper just read the results obtained are at best approximate. For example: The ratio of air to water does not represent with any degree of accuracy what actually takes place in the tower. The wind velocity varies continuously and the quantity of air supplied would change correspondingly. In the same way the quantity of water would not be constant when you consider that in any vertical section of the tower this quantity might be smaller near the bottom than at the top, due to water being carried off by wind.

A statement made by Mr. Horne regarding the effect of cooling surface in the tower may be misleading. The amount of surface, and particularly the design of this surface, are important factors in the cooling results obtained. While the contact of the air with the water is of prime importance, the shape of the cooling surface units and their arrangement in the tower are essential to the proper distribution and exposure of the water to the air passing through the tower.

I appreciate the difficulty of trying to establish a theory of cooling towers which would be applicable in all cases. The proper design is that which is the result of practical experience.

Edwin Burhorn—I regret very much that the authors were unable to complete their theory of cooling towers, which they started to formulate in their paper read before the Society at the annual meeting of 1914. In that paper the authors stated that they expected to present supplemental data in more elaborate and complete form with theoretical comparisons. I do not find that these data have been supplied in the 1915 paper and therefore look forward to this information possibly at our next annual meeting.

The authors stated in their first paper that they had not as yet succeeded in developing a general formula, but that their next efforts would be directed to that end. Being particularly interested in the final solution of the problem, I will await the submission of the general formula with interest. Until that time, however, it is not possible to offer any reasonable criticism as to the accuracy or otherwise of the authors' results. The present paper, however, seems to leave considerable doubt whether the authors will succeed as they state on page 8, "To completely prove it, will require experiments with a tower in *which all the four variables*, referred to under theory, can be exactly measured. This *is not possible* with the atmospheric tower we have so far used in this investigation." They further state that in this tower the distribution of the water is such that the total exposed surface cannot be accurately measured and on this account the cooling surface "defied their attempts to estimate." Apparently, therefore, the authors have given up the theory as applied to an atmospheric tower and suggest that "if possible they will continue their experiments with a forced draft tower if facilities are provided for accurately measuring the air supply, etc."

The authors also state that the results of these trials "when made" will be communicated to the Society for further consideration.

It seems to me that a theory that will apply to a forced draft tower will not apply to an atmospheric tower or a natural draft tower, and different formulæ would have to be prepared so as to apply to each particular case. I would suggest to the authors, therefore, that if possible, they complete their theory on the atmospheric tower without combining same with a forced draft tower. If they can also experiment on the forced draft tower and give us reliable data on this type of apparatus, they will certainly be of interest.

Willis H. Carrier—There are certain suggestions which I would like to make with reference to the cooling tower data obtained by Messrs. Horne and Coffey, and particularly with reference to the results of their experimental work in determining the rate of evaporation from a wetted surface. In the past I have done some work along the same line, not in actual cooling towers, but experimental work of a nature similar to that carried on by Messrs. Horne and Coffey, to determine the rate of evaporation occurring from a wetted surface under various atmospheric conditions. This experimental work, of course, bears directly on the cooling tower problem.

At the start, I wish to state that Messrs. Horne and Coffey have undertaken a very intricate and at the same time very interesting problem. I believe that the data which they have already obtained and which they are proposing to obtain in future experimental work with reference to the cooling tower will be of the greatest value to the engineering profession in arriving at some standard basis for guarantees in cooling tower performance.

The principal criticism which I have to make is in reference to their use and determination of the constant Theta (θ), which I note they have attempted to determine by means of the exceedingly ingenious arrangement described in their paper, and which consisted in blowing atmospheric air over a lozenge-shaped container partially filled with water and covered with absorbent wicking kept moist by the water drawn by capillary attraction from the interior of the vessel. They were correct in their observation that the temperature in the container was reduced to the wet bulb temperature of the air blown over the evaporating surface of the vessel and remained at this temperature regardless of the increase in the air velocity. This is a well-known psychrometric principle and obviously the temperature must be that of the wet bulb, because the wet bulb temperature is maintained in exactly the same manner, and the size and contents of the wet bulb member has no bearing upon the temperature obtained. However, the further assumption that since the temperature is constant for a given psychrometric condition of the air, the amount of heat transferred and the rate of evaporation is constant regardless of the velocity of the air, is incorrect. They are correct in the statement that the inflow of sensible heat from the air is exactly counterbalanced by the constant outflow of latent heat in the form of water vapor from the evaporating surface. In other words, while the transfer of sensible heat always equals the transfer of latent heat, both increase or decrease simultaneously with the air velocity over the evaporating surface. This would be expected in theory and has been demonstrated conclusively through experiments. The theory governing this wet bulb temperature and the reason for its being substantially constant was given by the writer in a paper entitled, "Rational Psychrometric Formulae" presented before the American Society of Mechanical Engineers in December, 1914, and will be found in the proceedings of that Society, together with discussions which may be said to fairly substantiate the conclusions reached. Further than this, I did some experimental work in the autumn of 1914 which bears directly on this problem, and Messrs. Horne and Coffey may find this data of value in their research.

The object of these experiments was to determine the rate of evaporation occurring from a free water surface at any definite temperature when subjected to a current of air of any intensity and under any known psychrometric conditions. In these experiments a small rectangular copper tank, measuring approximately 5 x 5 x 1½ inches deep, was filled nearly to the brim with water at an initial temperature of approximately 140 degrees. This tank was then placed in a relatively large rectangular air duct in such a manner as to avoid eddies and to insure, as nearly as possible, a parallel flow over the surface. The intensity of this air current could be varied and maintained constant at will, by means of a variable speed, motor driven fan. The velocity of this air current was accurately measured in every case by means of a Pitot tube, and an especially sensitive gauge capable of indicating very minute changes in velocity. A thermometer was inserted in the water and slowly stirred so as to obtain a uniform distribution of temperature as well as a correct average indication thereof. The time required for each interval of five degrees temperature drop was accurately noted and recorded by the observer and the experiment was continued until the temperature of the water practically reached that of the surrounding air. The experiment was conducted in a room under automatic temperature and humidity control, so that the psychrometric conditions could be maintained absolutely constant for each experiment and varied at will.

Preliminary radiation tests were made under exactly similar conditions and with the same weight of water with the exception that the container was covered so that no evaporation could occur. This enabled us to plot the radiation curve, giving the B.t.u. transmitted by radiation for various temperature conditions and air velocities. It also enabled us to correct the experiments for radiation losses in every case, and to obtain the curve of the true rate of evaporation. The B.t.u. transmitted during any given interval of time was measured by the drop in water temperature times the weight of the water used in the experiment, together with the water equivalent of the container.

The results obtained in these experiments were somewhat surprising to me, as I had been led to expect from experiments with the counterflow type of cooling tower that the rate of evaporation would be determined by the difference between the wet bulb temperature of the air and the temperature of the water entering. A great many experiments conducted with different air conditions and at different velocities, however, showed in this case that for any given velocity of air the B.t.u. transmitted by evaporation was invariably and exactly proportional to the difference between the vapor pressure of the liquid corresponding to its temperature, and the vapor pressure in the air corresponding to its dew point temperature.

It was also found that the rate of evaporation, i.e., the B.t.u. transmitted by evaporation, increased very rapidly with the air velocity. In the still air we had an evaporation corresponding to approximately 95 B.t.u. per square foot per hour for each inch (mercury) difference in vapor pressure between the water and the surrounding air, while at a velocity of 1000 feet per minute we had a rate of transmission of 520 B.t.u. per square foot per hour per inch difference in vapor pressure.

On Plate 6, I have plotted the results obtained in still air and at air velocities of 600, 1000, 1200 and 1400 feet per minute respectively and corresponding to one inch difference in vapor pressure. These were results obtained in a number of experiments over a wide range of differences in vapor pressure. This plot apparently gave a straight line as shown. We see from this that the equation for the rate of evaporation expressed in terms of B.t.u. and the velocity in feet per minute may be written in the form of $H=E\ (a+bv)$ $(e_w—e_a)$, where H is the B.t.u. per square foot per hour transmitted by evaporation, v is the velocity of the air over the surface in feet per minute, e_w is the vapor pressure of the water, e_a is the vapor pressure in the air, $a + b$ were experimental constants in our test. The values of the constants found were, $a = 95$, $b = 0.425$, when v is expressed in feet per minute, so that the equation for the rate of evaporation from a water surface at any temperature by air at any velocity and at any vapor pressure is $H = (95 + 0.425\,v)$ $(e_w—e_a)$.

I call attention to the fact that this equation is exactly of the same form as that given by some older investigators for evaporation from a water surface. I further call attention to the fact that this equation also holds true in the case where the water is at the wet bulb temperature, as in the case of the experiments of Messrs. Horne and Coffey.

It will be understood that this curve and equation for the rate of evaporation is the instantaneous rate corresponding to a given air temperature and a given water temperature, while in actual practice in a cooling tower we have an average rate of evaporation corresponding to an average or effective difference in vapor pressures between the water at a constantly decreasing vapor pressure,

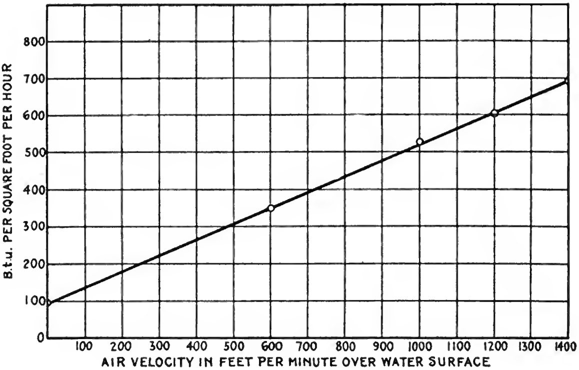

Plate 6

and the air at a constantly increasing vapor pressure as it passes over the wetted surfaces. For instance, in a tower of the type discussed by Messrs. Horne and Coffey, we have water entering the top at 90 degrees, leaving the bottom at 75 degrees, while we may have the outside entering air at a dew point corresponding to 60 degrees, and the wet bulb temperature corresponding possibly to 67 degrees, leaving the top of the tower at a dew point of possibly 80 degrees and leaving the bottom of the tower at a dew point of possibly 70 degrees. In a cooling tower of the counterflow type, this calculation would be somewhat less complex, especially if the quantity of air handled was definitely known. For instance, we might assume that a tower be so designed that the water could enter at 95 degrees in the top and leave the bottom at 75 degrees, with the air entering at a 60 degree dew point and at a 67 degree wet bulb temperature at the bottom, and leaving the top at 85 degrees saturated. The difference in vapor pressures at the bottom of the tower would then be 0.873—0.620 = 0.253 inches mercury, while at the top we would have a vapor pressure difference 1.42—1.03 = 0.39 inches. The mean effective vapor pressure difference would be therefore approximately 0.31 inches mercury, or slightly less than the numerical average.

If we may assume an effective velocity over the wetted surfaces in this type of tower of, say, 250 feet per minute, we will then have (by Plate 6) a cooling effect of 200 B.t.u. per square feet per hour times 0.31, or a cooling effect of about 60 B.t.u. per square foot per hour average throughout the tower. This will give the pounds of water which will be cooled per hour through a 15 degree range by 1 square foot of effective surface, as 60 ÷ 15 = 4 lbs. If the velocity were increased from 250 to 720, the rate of cooling would be twice as great, i.e., 120 B.t.u. per square feet per hour, and 1 square foot of surface would cool 8 pounds of water per hour through the same range. This example will give approximately a method to be followed out in applying the above data to cooling towers.

*Messrs. Horne and Coffey**—Taking up Mr. Lasker's points seriatim:

We quite agree that a cooling tower theory is of little value unless it can be made the basis of practical work, and this end we are endeavoring to attain, as stated in our paper. Our papers are merely to report progress, and obtain the benefit of the members' criticism of the steps in an investigation we know to be of considerable interest to the profession, but which may yet take considerable time to complete.

Regarding the air measurements, would say we have sought to minimize errors by using recording anemometers.

The water losses by evaporation and windage, except at high wind velocities, would hardly exceed one per cent, which we think introduces no error of magnitude.

The remarks about arrangement of cooling surfaces and distribution of water are quite true, but the mechanical features of a tower do not effect the law of heat transfer, which is what we are investigating.

Replying to Mr. Burhorn, he does not regret more than we do our inability to complete our investigation and formulate our conclusions in time for the 1915 meeting. Some of the difficulties we met with are mentioned in our paper.

Regarding the final proof of our theory with a forced draft tower, we evidently have not clearly stated the proposition. It is, of course, true that if the four variables in our final equations are known, and the calculated results agree with those actually obtained in practice, the theory must be admitted to be correct, and in no other way that we know of can it be directly proved. The total effective cooling surface is one of the variables, and as explained in our paper, cannot be measured in an atmospheric tower. If then we can establish the general theory with a forced draft tower in which all the variables, including the cooling surface, can be measured, we are in a position to apply the theory to atmospheric towers in the following manner: In a given atmospheric tower three of the four variables being known, and also the results in temperature reduction, we can, by putting these quantities in the final equations, solve for the unknown variable the effective cooling surface. In this way constants can be obtained giving the actual effective cooling surface in atmospheric towers for various drops between decks, at different rates of flow, and for various forms of drip bars. Once these constants are developed, they can be used to predict results under variable atmospheric conditions and rates of flow, and also to establish criterions of efficiency.

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* Authors' closure under the rules.

We express to Mr. Carrier our very great appreciation of his able and painstaking discussion, which will be of the greatest service to us in our work upon this intricate and baffling problem; particularly in concentrating our attention on the effect of variable air velocity upon the value of the constant Theta (θ) which we were in error in assuming to be independent of this variable after the wet bulb temperature was attained.

Since presenting our paper we have made, at Mr. Carrier's suggestion, a few random tests with the experimental apparatus shown in Figs. 2 and 3 of our present paper, after replacing the constant speed fan, *F*, by one of variable speed, and measuring the corresponding air velocities with a ring anemometer suspended in front of the evaporator, *A*. The observations were taken with wet bulbs ranging from 46 degrees to 55 degrees, and dry bulbs from 58 degrees to 74 degrees. The resulting values of θ are shown on the accompanying Plate 7, and indicate this value to vary directly as the air velocity. We will supplement and check these experiments later on, but we consider them conclusive in demonstrating the effect of air velocity upon the value of (θ).

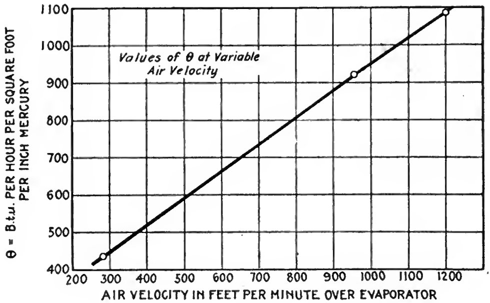

Plate 7

The equation $H = E\ (a + b\,v)\ (e_w - e_a)$, proposed by Mr. Carrier to express the heat transfer in a cooling tower reduces to $H = a\,(e_w - e_a)\,(1 + \frac{b}{a}v)$, which is equivalent to the general form of the equation expressing Dalton's law.

$$E = C\,(e_s - e_d)\,(1 + A\,W)$$

given in our first paper, A. S. R. E. Journal, November, 1914, page 79, in which

E = evaporation (weight or volume).
e_s = vapor pressure (inches of mercury) corresponding to water temperature.

e_d = vapor pressure (inches of mercury) corresponding to dewpoint temperature.
W = wind velocity.
A and C = constants.

This equation is quite old and has been used to frame numerous experiments by different investigators; but it fails to express the true conditions, as shown by the wide divergence in the constants. We quote from a paper by Prof. F. H. Bigelow, *Monthly Weather Review*, July, 1907, "There have been several important and extensive researches into evaporation phenomena such as exist over lakes and reservoirs in different climates, wherein attempts have been made to arrive at a general formula to express the results of the observations upon the amount of evaporation, in terms of several simple arguments."

He then gives the author's and the four following formulae derived from extensive investigations. We are personally acquainted with the Fitzgerald investigation, which was most elaborate and exhaustive, including both laboratory and field observations in great number. His paper, read before the American Society of Civil Engineers in 1886 should be carefully studied by those interested in the subject.

The formulæ cited by Prof. Bigelow are:

1. Abbassia (Egypt) $E|\text{mm/month} = 8.8\ (e_w - e_d)\ (1 + 0.0239k)$, where k = wind in kilometers per hour.
2. Fitzgerald (Boston, 1876–87) $E|\text{inch/day} = 0.3984\ (e_w - e_d)\ (1 + 0.0208\ W)$, where W = wind movement per day.
3. Carpenter (Fort Collins, 1887) $E|\text{inch/day} = 0.3868\ (e_w - e_d)\ (1 + 0.0025\ W)$.
4. Stelling (Russia, 1875–82) $E|\text{mm/2 hrs} = 0.0702\ \Sigma\ (e_w - e_d + 0.00319\ \Sigma\ (e_w - e_d)\ W$, where $\Sigma\ (e_w - e_d)$ = sum of twelve readings taken every two hours.

Prof. Bigelow then says:

All are in form Dalton's law,

$$E = A\ (e_w - e_d)\ (1 + C\ k)$$

Reduced to common form, standard units, and a specific example, the paper states these formulae give the following evaporation in millimeters per hour:

1. $E|\text{mm/hour}$ = 0.1337, or relatively 100
2. " = 0.2162 " " 162
3. " = 0.1504 " " 113
4. " = 0.3495 " " 261

The last column compares the results more clearly. We have taken the lowest at 100 and the others proportional thereto.

The professor then comments as follows:

"The agreement is so unsatisfactory as to suggest that the formulae have not a comprehensive form and that the so-called constants determined empirically are in reality variable to a considerable extent."

The unreliable results so far obtained in following the law of

Dalton applied to evaporation, and the consequently obvious errors that would be introduced if it were applied to the heat cycle of a cooling tower, were what induced us to undertake our present investigation in the hope of developing something better.

A further objection to using the Dalton equation in cooling-tower calculations is its failure to express the flow of sensible heat, which is a very important item at water temperatures high above the air temperature, when the heat transfer is from water to air; and again at water temperatures below the air temperature when the heat transfer reverses. In this respect our general equation (3), page 86 of A. S. R. E. JOURNAL, November, 1914, is superior, as it expresses both the latent and the sensible heat flow.

One more point in which we disagree with Mr. Carrier concerns his opinion that an approximate mean of the air-water vapor pressure differences on entering and leaving the tower would express the mean effective difference for the cycle. An inspection of Plate 4, page 88, A. S. R. E. JOURNAL, November, 1914, our first paper shows that the air and water curves expressing the differences E at all stages of the passage do not approximate straight lines, and that therefore the mean difference will not even approximate the mean of the extremes. Additional examples of these curves are shown on Plate 3 of our present paper. In fact one of our chief objects at present is to find a simple, workable expression that will give this mean difference. It is true, Mr. Carrier refers to actual vapor pressure differences alone, while our curves express a somewhat larger difference, due to the combined sensible heat factor; nevertheless the curve forms remain substantially similar and the argument holds good.

In closing, we wish again to thank Mr. Carrier for what we term his constructive criticism, which in the general search for truth is of vital value.

AMMONIA CONDENSER DATA*

By Henry Torrance, New York, N. Y.

DOUBLE PIPE CONDENSER TESTS

At the Tenth Annual Meeting of this Society in 1914 the writer presented test data showing that a double pipe ammonia condenser with the same tonnage and water supply gave the same results whether the water and ammonia respectively and countercurrently ran up-hill or down-hill. These data were published in the A. S. R. E. Journal for May, 1915.

At that time it was said the test data were not complete, as the anhydrous ammonia circulated was not weighed. This was true, but the tonnage was the same, hence the quantity of liquid must have been about the same.

These tests showed that in a double pipe condenser it does not matter whether the ammonia runs uphill or downhill (water to run countercurrent) and that "flooding" makes no gain.

FLOODED ATMOSPHERIC CONDENSERS

This past summer further experiments have been made on the same test plant, with the same machinery, on atmospheric condensers. A standard flooded condenser of the injector type, Fig. 1, was purchased from the manufacturer and tested. There was inserted a gate valve on the fourth run of pipe from the bottom, so that when closed the condesner became the ordinary type in use for years and in which the gas first went through three runs of pipe at the bottom, so that the hot gas would first be cooled by waste water, then it entered the top of the coil as usual and ran down hill as usual.

A number of tests were run and the *results shown in Table I appear a little better with the valve closed,* considering that the surface was reduced. Fig. 2 shows the valve removed.

In Fig. 3 are plotted the weights of ammonia condensed per square feet per hour per 1 degree Fahrenheit mean temperature difference against the total amount of ammonia condensed per hour.

THEORY OF CONDENSATION

Fred Ophuls first pointed out to the writer that a vapor will condense best when it actually touches a dry surface and that any liquid present will serve to keep the vapor away from the surface.

The writer believes that:

In a condesner where vapor is condensed the vapor should touch the pipe surface.

In a brine cooler where liquid is evaporated the liquid should touch the pipe surface.

A condenser should run empty as far as possible.

A brine cooler should run flooded as far as possible.

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* Read at Eleventh Annual Meeting, New York, N. Y., December 6, 7 and 8, 1915.

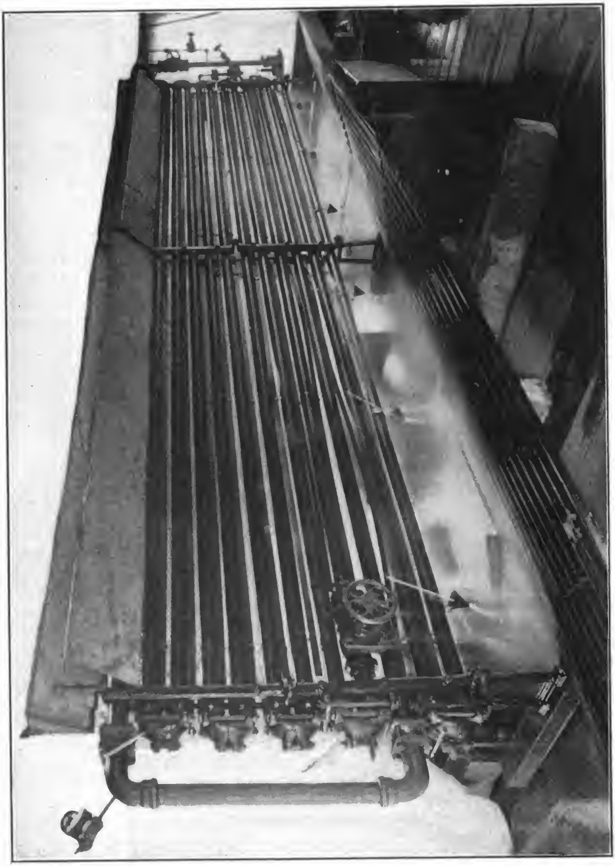

Fig. 1

TABLE I

	Date	Sq. Ft. of Condensing Surface.	Tons of of Actual Refrig.	Sq. Ft. Cond. Surf. per Ton of Actual Ref.	Head. Press. in Lbs. per Sq. In.	Temp. Due to Head. Press., °F.	Temp. of Liquid from Cond. °F.	CONDENSER WATER		
								Inlet °F.	Outlet °F.	Range °F.
With Injector	Nov. 18, 1915	150	10.4	14.4	128.4	75.8	72.1	59.7	69.9	10.2
Without Injector	Nov. 19, 1915	137	9.5	14.4	127.3	75.3	70.4	59.6	69.4	9.8
With Injector	Nov. 20, 1915	150	13.5	11.1	121.5	72.9	70.0	59.8	66.5	6.7
Without Injector	Nov. 22, 1915	137	13.1	10.4	123.5	73.7	69.9	60.2	67.2	7.0
With Injector	Nov. 23, 1915	150	22.1	6.8	133.7	77.9	73.0	59.8	68.2	8.4
Without Injector	Nov. 24, 1915	137	27.4	5.0	135.3	78.5	74.2	59.7	68.1	8.4
With Injector	Nov. 29, 1915	150	15.4	9.74	125.1	74.4	70.9	59.9	67.2	7.3
With Injector	Nov. 30, 1915	150	27.8	5.4	147.0	82.9	78.9	60.8	72.3	11.5
Without Injector	Nov. 30, 1915	137	28.5	4.81	144.0	81.8	77.9	60.1	71.8	11.7
With Injector	Dec. 2, 1915	150	35.9	4.1	153.8	85.4	80.9	60.6	72.1	11.5
Without Injector	Dec. 2, 1915	137	36.5	3.75	151.0	85.5	80.6	60.0	71.2	11.3

	Date	MEAN TEMPERATURE DIFFERENCE BETWEEN TEMPERATURE DUE TO HEAD PRESSURE AND CONDENSER WATER TEMPERATURES, °F.		Weight of NH_3 Condensed per Hour, Pounds.	Weight of NH_3 Condensed per Hour per Square Foot Condenser Surface, Pounds.	Weight of NH_3 Condensed per Hour per Square Foot of Condensing Surface per 1°F. Log. Mean Temp. Difference, Pounds.	Temperature Gas to Condenser, °F.	Gallons Water Per Minute.
		Arithmetical.	Logarithmic.					
With Injector	Nov. 18, 1915	11.0	10.1	265.0	1.77	0.175	101.0	25.5
Without Injector	Nov. 19, 1915	10.8	9.95	256.9	1.87	0.188	99.2	25.5
With Injector	Nov. 20, 1915	9.8	9.3	339.9	2.37	0.243	100.9	49.9
Without Injector	Nov. 22, 1915	10.0	9.5	339.4	2.48	0.260	99.5	49.9
With Injector	Nov. 23, 1915	13.9	13.3	593.2	3.95	0.297	113.7	74.6
Without Injector	Nov. 24, 1915	14.6	14.3	662.1	4.80	0.337	114.6	74.9
With Injector	Nov. 29, 1915	10.8	10.5	360.0	2.4	0.228	101.8	52.1
With Injector	Nov. 30, 1915	16.3	15.7	751.0	5.0	0.318	124.3	74.5
Without Injector	Nov. 30, 1915	15.8	15.2	758.0	5.53	0.364	123.1	74.9
With Injector	Dec. 2, 1915	19.0	18.5	992.0	6.61	0.357	139.0	102.4
Without Injector	Dec. 2, 1915	19.9	19.9	1012.0	7.39	0.379	135.0	103.3

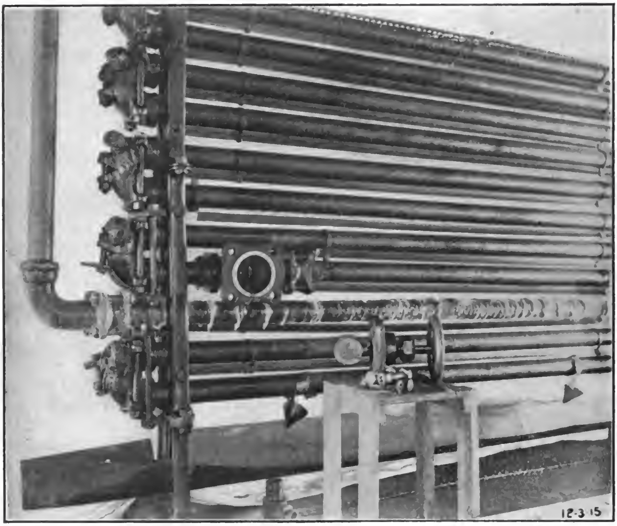
12-3-15

Fig. 2

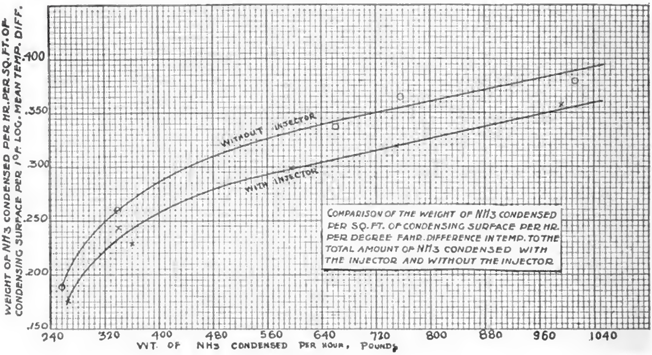
WEIGHT OF NH3 CONDENSED PER HR. PER SQ. FT. OF CONDENSING SURFACE PER 1°F. LOG. MEAN TEMP. DIFF.
.400
.350
.300
.250
.200
.150
240
320
400
480
560
640
720
800
880
960
1040
WT. OF NH3 CONDENSED PER HOUR, POUNDS
WITHOUT INJECTOR
WITH INJECTOR
COMPARISON OF THE WEIGHT OF NH3 CONDENSED PER SQ. FT. OF CONDENSING SURFACE PER HR. PER DEGREE FAHR. DIFFERENCE IN TEMP. TO THE TOTAL AMOUNT OF NH3 CONDENSED WITH THE INJECTOR AND WITHOUT THE INJECTOR

Fig. 3

DISCUSSION

President Doelling.—Was the gate valve tight?

Mr. Torrance.—As far as we knew. We thought a gate valve would give a fairer test.

Louis Block.—What is that on the third pipe from the bottom in Fig. 2.

Mr. Torrance.—Ice; when the condenser was shut down and the ammonia drawn off there was a little freezing.

Louis Block.—How did you arrive at the weight of the ammonia?

Mr. Torrance.—It was weighed in receivers, first filling one and then the other. Our Mr. Ewer will be able to explain that.

Louis Block.—Why do you figure that you cut out one pipe when you really cut out two pipes?

Mr. Torrance.—I did not wish to exaggerate it. I did not think it was fair to say that two pipes were cut out; that is a question of opinion.

Louis Block.—I think not. Now, Mr. Torrance, you say that with the valve closed, and consequently putting two pipes out of commission, you are doing more work than with the valve open and those two pipes in commission, and since I cannot suppose that the injection of cold liquid into a gas, which has parted with its superheat by that time and is just about ready to condense, can do any possible harm, would it be logic to conclude that if you took two more pipes out you would still do more work?

Mr. Torrance.—We got better results in spite of the fact that there was one less pipe because there was no liquid injection.

Louis Block.—Then take out two more pipes and you get still better results?

Mr. Torrance.—This was a comparative test between two types of condensers, each having the same surface.

Louis Block.—I say it cannot possibly do any harm to inject a cold liquid, which is colder than the gas, into a gas which is just about ready to condense.

Mr. Torrance.—The condenser with the injector took more surface to do the same work.

Louis Block.—You have made some sort of mistake in your test, which I cannot fathom just now, but it seems impossible and this theory of cooling the gas in contact with dry surfaces has long been exploded; that is simply the theory which steam condenser builders or manufacturers have followed and they are mistaken.

Mr. Torrance.—Mr. Block, we are talking about a vapor, not a gas.

V. R. H. Greene.—Inasmuch as I co-operated with Mr. Torrance in this test, I would like to show that Mr. Block is wrong. As Mr. Block very truly said, it has been well known in steam condenser practice for a great many years that to keep the cooling surfaces as dry as possible produces a maximum rate of heat transfer, and that has been looked into by so many eminent authorities that I do not think it is up to this Society to question the statement. In the still more efficient surface steam condensers recently brought out, and this has also been brought before the Society before, baffle plates have been cast on the side of the cast iron shell to receive the liquid as fast as condensed and take it away.

Louis Block.—How are those baffle plates arranged?

V. R. H. Greene.—Cast on the side of the shell.

Louis Block.—To hold the liquid in?

V. R. H. Greene.—No, to take it out.

Louis Block.—And pour it on the tubes again?

V. R. H. Greene.—No, sir; take it away as fast as condensed.

Louis Block.—Are the baffles very close to the tubes?

V. R H. Greene.—No, the whole space is filled with tubes with the baffles cast along the side, but the idea of the baffles is to get rid of the liquid. The reason is this; while it has been stated before this society that the gas to water transfer is the poorest possible transfer we can get, the reason being that when a gas is in contact with its own molecules of gas there is no way that the heat can pass through the pipe surface into the water outside of the pipe surface except by conduction from one molecule to the other; we must remember that ammonia is a vapor and not a gas and the foregoing statement does not apply. When a vapor, particularly near its liquefaction point, is in contact with a pipe surface with water on the outside it collapses and should be drained away to allow another molecule of vapor to get in contact with that same surface. The probable reason for the increase of the rate of heat transfer by increasing the velocity, which was brought out at one of the previous meetings by Mr. Ophuls, is that high vapor velocities keep the liquid film on the condensing surface thin. so that the oncoming molecules of vapor may strike upon that surface and there be condensed. I would like to show you why this seems to work out in this particular test. We used a standard condenser where the vapor enters at the top and the liquid drains out at the bottom. We knew the relative velocity of the vapor in the pipes.

Louis Block.—How did you ascertain the relative velocity?

V. R. H. Greene.—In this old style of condenser, with the vapor going in at the top and the liquid coming out at the bottom, we found the weight of ammonia circulated per hour, and so figured the velocity of the vapor at the entrance to the condenser. We assumed in this paper that the vapor entered the top at maximum velocity and came out at the bottom at zero velocity, being all condensed at the outlet. At the same time there could be but little liquid present, because the vapor carried the liquid down with it. We plotted a series of curves, the ordinates being the surface and the abscissae being the velocity. It was a simple matter to get the average vapor velocity through the condenser and then with the aid of Mollier's equation we figured out what the rate of heat transfer was on the original old type condenser. The next move to make was to reverse this condenser and have the gas go in at the bottom and the condensed liquid come out at the top. Here we assumed that the liquid began to build up in the upper 9 or 10 pipes. Manifestly no liquid could be in the bottom pipes, because our test showed that two or three of the bottom pipes were required to take out the superheat, and any liquid at this point would be above its liquefaction temperature. Somewhere higher up in the condenser the ammonia vapor began to condense and accumulate, and when an excess of ammonia collected in these pipes the vapor carried it to the next pipe higher and so on up until liquid was discharged at the top pipe. For this type of condenser we obtained a curve showing that the velocity of the vapor through the pipes was higher than in the old type of condenser. This is brought about presumably by the presence of the liquid reducing the effective area of the pipe so that the vapor must travel over the top of the liquid and in this manner a higher velocity is maintained and a higher rate of heat transfer obtained than in the old type with the top feed and the bottom liquid drain. Now the next point was to see if this theory was applicable and see if it was not true, as is true in steam condenser pratice, that the rate of heat transfer in the condenser is a function of the velocity of the vapor and not of the liquid which may be present in the condenser. From the data which Mr. Torrance has presented we obtained some very interesting information, shown in Tables II and III. The figures in Table II were obtained with the valve closed, which means that the vapor entered the bottom and passed up the first three pipes, for convenience called the superheat pipes, then went up to the top pipe, and from there down and out of the drip connection, which made it the regular, old time condenser used before the introduction of the ejector. In all these tests, the liquid ammonia was weighed, the brine was weighed, the temperature ranges taken and the heat balance checked up within less than 5 per cent, which I consider very close for work of this character. Knowing the effective cooling surface of this condenser, and not making any allowances for one less pipe, as Mr. Torrance did, I took the cooling surface at 150 square feet. Knowing the velocity of the vapor at entrance, its mean square root speed and the rate of heat transfer as shown by Table II we can calculate the total heat. Table III shows the heat rejected with the valve open. The initial velocity was taken as equal to the velocity of the gas through the ejector orifice, which was somewhat less than one inch in diameter. We knew how much vapor went through the orifice, because we had determined the weight of ammonia in the same manner as in the previous case. We calculated the velocity through the ejector nozzle, which we took out and carefully calibrated, and found the velocity very high. The only assumption we had to make in the calculation, which we did intentionally, was to assume that because this is a flooded condenser, and because liquid does come down through the pipes above and enters the annular space of the ejector where the vapor enters, that the liquid lay in the pipes, filling them partly full of liquid and forcing the vapor to travel at a proportionately high velocity through the space between the surface of the liquid and the inside of the pipe. We figured this way so as to assume that the velocity throughout the condenser was practically constant until the end, and in that way obtained a rate of heat transfer

TABLE II.—ATMOSPHERIC CONDENSER

	Absorption Plant			
Dates of Test, 1915..................	Dec. 2.	Nov. 30	Nov. 22	Nov. 24
Rate of Heat Transfer, k_m............	235.	226.9	170.1	212.5
Heat Rejected in B.t.u. per hour......	535,800	399,960	179,280	350,520
Initial Velocity at Condenser in ft. per second, v_1..........................	22.8	18.2	9.14	16.67
Mean Sq. Root of Speed, $v_m = {}^4/_9 v_1$	10.1	8.1	4.07	7.42
$k_m = a\sqrt{v_m}$	73.9	79.9	84.3	78.2
Then $a = \frac{k}{\sqrt{v_m}}$..................	....	Average 79.1		....

TABLE III.—ATMOSPHERIC FLOODED CONDENSER

	Absorption Plant				Comp. Plant
Dates of Test, 1915..........	Dec. 2	Nov. 30	Nov. 29	Nov. 20	Oct. 20
Rate Heat Transfer, k_m, as Calculated from $k_m = 79.1\sqrt{v_m}$.	831.	733.	541.	531.	649.
Heat Rejected in B.t.u. per hr.	529,200	396,480	189,720	179,940	362,400
Initial Velocity at Condenser in ft. per sec., v_1..........	110.2	86.	46.9	45.1	67.2
Mean Logarithmic Temp. Difference between Water and Vapor in Condenser*.....	13.10	11.90	7.32	6.56	9.82
Effective Cooling Surface....	48.6	45.4	47.8	51.7	56.8
a as taken from Tests on Atmospheric Condenser.....	....	Average 79.1		....	....

* Difference in the mean temperature differences found for the same tests in Tables I and III are due to the fact that the former are based on the temperature due to the discharge pressure at entrance to the condenser and the latter on the actual liquefaction temperature in the condenser.

which ran as high as 831 B.t.u. on test of Dec. 2, with tremendously high vapor velocity going through the pipes to produce this rate of heat transfer. Now the temperature difference, that is, the logarithmic mean temperature difference was, 13.1. If we know the rate of heat transfer and we know the logarithmic mean temperature difference and the total B.t.u. rejected, it is simple to find the effective cooling surface of the condenser. The effective cooling surface is as shown in Table III.

Louis Block.—How do they happen to vary?

V. R. H. Greene.—They do not vary very much.

Louis Block.—Why do they vary at all?

V. R. H. Greene.—These are different tests on different days under different conditions.

Louis Block.—That does not make any difference; the condenser does not change from day to day.

V. R. H. Greene.—The tonnage changes.

Louis Block.—You say the effective cooling surface; you cannot change the cooling surface unless you let one part of the condenser run dry.

V. R. H. Greene.—That is the beauty of the whole test. It shows that you have 150 square feet of cooling surface in that condenser, of which only 48.4 worked; the balance must have been partially filled with liquid—enough to reduce the effective surface.

Louis Block.—You do not suppose the liquid can lie there and not flow out?

V. R. H. Greene.—I surely do.

Louis Block.—When you get behind it with a velocity of 110.2 feet per second.

V. R. H. Greene.—That only stirs up the surface.

Louis Block.—You have never looked in there?

V. R. H. Greene.—I have not.

L. Williams.—What is the 831?

V. R. H. Greene.—The rate of heat transfer.

L. Williams.—What surface did you use in calculating that rate of heat transfer?

V. R. H. Greene.—I took the rate of heat transfer from the same equation that I used all the way through and found out the average velocity in this condenser.

L. Williams.—Then the only chance you take is on the assumed velocity through the jet?

V. R. H. Greene.—That is true.

L. Williams.—If the gas, after passing through the jet slows down, then your deductions are wrong?

V. R. H. Greene.—Yes, and the only thing I can offer to support this deduction is that these figures stay the same for all tonnages.

L. Williams.—But your deduction remains about the same regardless, doesn't it? It varies between the tonnage?

V. R. H. Greene.—I have obtained all kinds of velocity, but the surface remains practically constant. This leads me to believe that the condenser pipes are probably half full of liquid, and consequently I have to change my temperature difference to do effective work.

L. Williams.—Then the same percentage of the condenser is effective regardless of the tonnage you are putting out?

V. R. H. Greene.—Yes, because the same amount of liquid lies in the bottom of all those pipes, which you would expect.

L. Williams.—I would expect that with a higher velocity it would carry the liquid over.

V. R. H. Greene.—No, it only stirs up the surface.

L. Williams.—The tendency would be to bank the liquid higher in the pipe?

V. R. H. Greene.—The pipes will stay as full as they can to permit the gas to pass over the top of it.

L. Williams.—But that fullness will vary with every different velocity?

V. R. H. Greene.—Only slightly.

L. Williams.—But it seems to me that that situation is not one on which you can base deductions of this nature; it is too far-fetched.

V. R. H. Greene.—Personally I feel that there is no question about it. We have advanced theories before to prove that the rate of heat transfer is governed more by the velocity of the vapor than by any other thing, and here are the actual facts. I am trying to satisfy these facts from an engineering standpoint based on theory, and this deduction comes close to satisfying the actual conditions.

L. Williams.—It seems to me that the reasons I have just advanced upset your fundamental deductions and would possibly make the last line in your tables all wrong.

V. R. H. Greene.—But that cannot affect the results; this is the work the condenser actually did; it is not based on theory.

L. Williams.—One of them must go.

V. R. H. Greene.—The only variable of the proposition is, does the velocity stay constant? You cannot question the facts because they prove themselves, since they form a heat balance.

Fred Ophuls.—I think that in this discussion we lose sight of the fact that the tests showed there was no difference between the two styles of condensers. The calculations we have offered here are only a means or attempt to check up the results that we have obtained and explain why the presence of liquid has no bearing on the operation of the condenser. We do not offer this as actual scientific data, but we do offer it as proof that even if the liquid lay in the condenser or the velocity kept constant or even if the velocity varied we can show that less surface than the whole surface of the condenser was effective in producing the tonnage. That is the only object of the paper, and in this discussion we must keep in mind the fact that from the 11 tests that were made, five with the valve closed and six with the valve opened, no difference at all was noticeable in the operation of the condenser.

Mr. Block said before that the theory that a vapor will condense more rapidly when brought in contact with a cool dry surface was exploded. This is not so. In steam condenser practice it is still regarded as the correct theory. There are a great many more eminent men engaged in steam engineering or have been in years gone by than one can find in the refrigerating industry, so we cannot discard their theories without a great deal of most scientific investigation. The tests I made at the De La Vergne Machine Company were made to analyse condenser performance. These results I presented last year before this Society. They show conclusively that there seems to be a change in the heat transfer due to vapor velocity. In the various types of condensers I analyzed, this change was certainly due to a decrease in the thickness of the liquid film on the pipes when the velocity of the vapor increases. The correctness of this theory you will readily admit if you will consider the difference between the action of the molecules of a gas and of a vapor on being cooled.

The removal of heat from a gas must take place either by conduction, convection or radiation or a combination of these. Only a small part of the molecules of the gas being cooled actually come into contact with the cooling surface, but in the case of a vapor, the action of the molecules are entirely different. As one molecule collapses, the succeeding molecule has an opportunity to come in contact with either the liquid covering the cooling surface or else with the surface itself, if it is dry enough, and therefore if one could, for instance, by rotating the cooling surface at tremendous speed, keep the film of liquid on the cooling surface very thin, one would probably be able to approach the actual conductivity of the metal as far as the heat transfer co-efficient is concerned.

Louis Block.—How do you know that high velocity keeps the surface clean in a condenser of this kind?

V. R. H. Greene.—That is just what we are trying to prove.

Louis Block.—You are only guessing.

V. R. H. Greene.—We are not guessing; these are the results of actual tests; nothing is assumed.

Fred Ophuls.—The idea is this; we used the five tests made on the condenser with the valves shut to find the constant *a* in Mollier's equation of the relation between the heat transfer co-efficient and the velocity of the vapor and with this data we figured the amount of effective cooling surface required to give the results obtained with the ejector type of condesner.

L. S. Morse.—These figures are very interesting but they do not give all of the test data. You will notice that Mr. Torrance has omitted the amount of water circulated. I would like to have him give approximately the amount of water circulated for those conditions.

Fred Ophuls.—You can get that from the temperature difference of the water.

L. S. Morse.—I would like to know if you have this data.

Mr. Torrance.—Mr. Ewer had those figures and also the temperature of the gas.

Fred Ophuls.—The amount of water showered over the condenser makes no difference, that I can see.

L. S. Morse.—In guaranteeing condenser capacity we are governed by temperature and amount of water, hence I would like to know how much water you used.

Fred Ophuls.—You can tell close enough by the temperature difference of the water, allowing 3 gallons a ton or something like that. It ranges from 6 to 3 gallons. If it is five degrees difference, it is about 6 gallons a ton.

V. R. H. Greene.—We are not criticising the flooded condenser; we obtained a tremendous rate of heat transfer, and what produces this is the question. Is it due to the presence of liquid or to the high vapor velocity?

L. S. Morse.—I am trying to find out how you arrived at 3 gallons per ton.

Mr. Torrance.—The water inlet temperature and the water outlet temperature are given in Table I; that is what affects the condenser. In the third test the quantity ran up a little, and the range was decreased.

L. S. Morse.—That is approximately 3 gallons then?

Mr. Torrance.—Yes, a ten degree range is about 3 gallons.

L. S. Morse.—Well, that is the question I wanted to bring up. You see that three gallons per minute of 60 degrees water was used, which amount is excessive for the temperature. I have no comparative data available, at least none here, but there is no question that this particular condenser, if properly operated as a flooded condenser, and showered with water uniformily distributed, should have shown a better capacity. This condenser, even with higher temperature water, should maintain 135 lbs. condensing pressure under the same tonnage, with 50 per cent less water.

V. R. H. Greene.—I purposely omitted from Table III a test which I ran on a compression machine with flooded condenser many miles away from Carbondale which was so nearly identical that there was very little difference. I will put them in the last column of Table III. We used only one gallon of water per ton per minute on the condenser. This was a compression machine; water circulated, 32 gallons a minute, the heat rejected in B. t. u. per hour was 362,400; the initial velocity at condenser in feet per second was 67.2; the rate of heat transfer, k_m, was equal to 649; the mean logarithmic temperature difference between water and vapor was equal to 9.82 and the square feet of effective cooling surface was 56.8. You see we had superheated vapor in this case. I left this test out because we had superheated vapor, whereas in the absorption machine the superheat is taken out at the rectifier; that makes a little variation, which I did not want to confuse with the other tests, but you see the general result is the same with only 56 out of 150 square feet effective.

L. S. Morse.—There must be something wrong about these figures, because they do not bear out the actual practice. Take 56 square feet; that was about four pipes.

V. R. H. Greene.—I say that out of 150 square feet of surface in the condesner tested in conjunction with the compression machine only 56 square feet was actually doing work.

L. S. Morse.—That is for cooling and condensation?

V. R. H. Greene.—Yes; the balance is filled with liquid, which keeps up the velocity.

L. S. Morse.—The liquid distributed over the whole condenser?

V. R. H. Greene.—Yes, larger at the bottom and less at the top.

A. Winborg.—I believe that Mr. Torrance would be disappointed if we could not possibly find something wrong with his data, because he sent us an abstract of his paper before we came to the meeting. Now, we only received some figures of the results obtained; for instance, one of the most important parts is the amount of water circulated in each test. Mr. Torrance thinks it is something like 3 gallons a minute per ton. If that was the case, he should have obtained a lower pressure than he did on a flooded injector condenser of the standard York type and, in fact, with the head pressure obtaining he should have obtained something like 40 per cent more tonnage than he gives in his results. Now there evidently was something absolutely wrong with the operation of that condenser. There is one thing in this test that is different from the tests run on the condenser in York, and that is the temperature of the discharge gas as it hit the condenser. Of course we had a compression plant and superheated gas; in this case it is an absorption plant and a dehydrator doing a good deal of the cooling work on the gas. Now, it is possible, and I might say probable, that the lower part of the condenser, or what we call the pre-cooler, was actually flooded in this case, that is, the gas gets in there nearly saturated and it is condensed in there while blowing up through the pre-cooler, through the liquid in the two pipes and also through the third pipe, and it is probable that in this case, whether the valve was open or closed, the effective part of the condenser would only be the three lower pipes and the riser, and that the upper part of the condenser was plugged with air and uncondensible gases; hence whether the valve was opened or closed, it would not make a whole lot of difference. I have no figures showing what the temperatures of the return bends were.

Fred Ophuls.—I would like to see some figures that substantiate Mr. Winborg's statement. As a matter of fact, the temperature of the vapor was taken and it was still superheated when it went into the injector.

Louis Block.—It is hard to believe, with that quantity of water, that the gas would still be superheated after having been through two pipes.

A. Winborg.—We obtained something like 40 per cent better results under those conditions.

V. R. H. Greene.—When this data was checked up, the ammonia weighed, the brine weighed and the temperature conditions kept, we obtained results that checked within five per cent.

A. Winborg.—You may have had uncondensable gases in your pipes.

V. R. H. Greene.—We watched that pretty closely.

Mr. Torrance.—I want to point out the surfaces per ton in Table I, 13.1 tons on this condenser figures 10.4 square feet per ton of actual refrigeration; which gives a head pressure of 123.5 lbs. with an outlet water temperature of 67 degrees F., which isn't bad; there could not be much air in the machine. We were very careful about the air, which we blew down into the absorber.

L. Williams.—Have you any definite knowledge whether the amount of liquid available for the jet was amply sufficient?

V. R. H. Greene.—No, except as indicated by thermometers.

L. Williams.—I mean the liquid receiver from which the jet draws its liquid; if liquid were not sufficient to follow up the jet action, one could imagine that the jet would have an intermittent action. Was there any means of getting at that?

V. R. H. Greene.—No, there is not, except the fact that we had thermometers on both sides of the jet.

L. Williams.—Was your liquid receiver more than the capacity of the jet?

V. R. H. Greene.—We had three liquid receivers; a big one at the outlet, then a weighing receiver and a terminal receiver.

L. Williams.—Was it not possible that there might have been a throttling action there? The jet action depends on the orifice, and its continuous operation depends on the liquid being able to follow the gas out of the orifice. There must have been some reason for the difference that, as I understand, you have got and others are getting, and it just occurred to me that it may be possibly in the liquid not being able to follow up and feed the injector.

V. R. H. Greene.—You recall the construction; gas enters at the bottom of three pipes and then passing upward strikes the jet then goes up to the top pipe and down nine pipes to feed the jet. At the eighth pipe is the outlet to the liquid receiver. On a single stand we carried tonnages as high as 40 tons, and the quantity of the liquid going through the condenser is pretty high and since we have only a small liquid outlet bleeder, and since the temperatures on both sides of the jet were so near alike there is no indication that hot gas was blowing through the jet.

L. Williams.—Even if that were the case and the jet were working intermittently, your thermometer would not show the intermittency that was taking place. The thermometer, therefore, would have to be discounted; there must be some difference; others get one result and you get another result.

We have a gentleman with us who has been trying an entirely different type of jet to any of these. I refer to Mr. Fairbanks. He showed me a type of jet on a very large scale which he is testing and in that jet he is getting a very close contact between the liquid and gas, he gets them well mixed up, and if he can give us any results, it might throw light on some of the differences which have cropped out.

V. R. H. Greene.—I would like to have his views.

G. T. Voorhees.—Professor Reed made a suggestion which may possibly give light on the whole question. The suggestion was what was the temperature of the liquid ammonia when it came in contact with the jet. It would look quite evident that if the temperature of that liquid ammonia was not below the condenser temperature due to that pressure, you would have lost the temperature of the injector. It seems to me that is the first inkling we get of what possibly might be the trouble.

Fred Ophuls.—How could the liquid exist under that pressure?

G. T. Voorhees.—It can exist at any temperature below that due to its pressure, if you cool it.

Mr. Torrance.—In other words, in order that the injector should work, the liquid must be under-cooled.

President Doelling.—The vapor or gas is condensed when it meets the liquid from the injector, but will it make an emulsion? You cannot inject too much liquid at one time. If you want to get a quick condensation, a special apparatus must be built and that apparatus must be very long. The principle of the jet apparatus is that the gas or vapor is condensed at the smallest point, at the throat. If that distance is not long enough or if the amount of gas is too great in comparison to the quantity of liquid, then it cannot be condensed. You must then make an apparatus that is long and it must bring in the liquid in a number of places to condense the vapor, just as you do in an ejector condenser.

L. Williams.—A number of jets give a better emulsion than a single jet.

President Doelling.—It takes a certain time, which is an element that comes in everywhere.

L. Williams.—Obviously, if there were any shortage of liquid, the inefficient jet would be more of a handicap still.

President Doelling.—The injector shown is a very poor injector as a jet apparatus; this apparatus, as far as I see, works only by impact like an air pump; it pushes the gas ahead; I don't think this device ever works as an injector.

L. Williams.—You lift the liquid, apparently.

Fred Ophuls.—The liquid comes down by gravity; it don't have to be lifted. The condenser is so constructed that it cannot help being flooded all the time. The liquid naturally goes into the injector first, before it ever gets into the receiver.

G. T. Voorhees.—I do not believe you have an injector; you have an ejector, which is a very different proposition. If the opening in the throat is smaller than the nozzle you have an injector, if it is larger you have an ejector.

Mr. Torrance.—The gas enters the bottom of the coil and goes up through two pipes at the bottom and through the injector and meets the liquid which drains down from the top where it draws in the liquid and carries it to the top of the coil. The injector is peculiarly constructed with a nozzle and combining tube and the diameter inside at the smallest gas opening is three quarters of an inch and it then flares out to about seven eighths of an inch. The liquid comes down all around the nozzle.

Louis Block.—How did you know that you got any liquid at the point of the injector?

Mr. Torrance.—I do not know; I don't think anybody knows.

Fred Ophuls.—A thermometer was placed there.

Louis Block.—The thermometer would not tell you whether there was any liquid.

President Doelling.—There could not be a vacuum, something must be there.

G. T. Voorhees.—What was the temperature of the liquid at the injector?

V. R. H. Greene.—Practically the same as the temperature of the liquid at the outlet.

Mr. Torrance.—I do not believe that much vapor condenses at the injector.

V. R. H. Greene.—You could never get the liquid at the injector undercooled enough to have the sensible heat of the liquid take out any latent heat.

Mr. Torrance.—I figure roughly that in order to have the cold liquid condense gas at the injector, that the liquid would have to circulate through the coil at the rate of over 100 gallons a minute, which is much more than the coil could handle, so it is absolutely out of the question to have all the vapor condense at the injector.

G. T. Voorhees.—I mean that it must be below the condensing temperature at the injector in order to be effective.

F. L. Fairbanks.—I think it would be hardly wise to inject anything into this discussion at this time, but I would like to see this discussion go on as it is going with a view to settling the question of the comparative efficiency of the flooded type condenser as compared with the older, or dry type.

The condenser to which Mr. Williams refers, which I have designed and have been experimenting with, is of the jet type, in which the principles of the well known jet steam condenser are utilized. This consists mainly of mingling directly and intimately a shower of liquid and the gas or steam to be condensed. This jet condenser for ammonia was designed for a capacity of 1,000 tons in 24 hours, and is only 24 inches in diameter and about six feet high over all, requiring for its installation an exceptionally small space. The cool anhydrous ammonia from the liquid coolers is pumped to the liquid jet in the condenser by a large aquà ammonia pump driven by steam, making it possible to spray through a special nozzle into the gas, coming from the compressors, 1 to 600 gallons per minute of the liquid. The gas and liquid are then brought more intimately into contact with each other by a series of baffles, and the resultant condensate flows by gravity through liquid coolers, which brings it down to within three to five degrees of the condensing water. From the liquid coolers it flows by gravity to the ammonia pump and thence to the jet of the condenser again.

The gas from the compressors flows into the top of the condenser through a 12 inch header and there are 12 thermometers tapped into the condenser so that the temperature is indicated very accurately, both before and after the gas and liquid are mingled. The condenser is so located on the gas header that it is only necessary to shut off the liquid jet to make it merely a fitting through which the gas flows to the liquid coolers and, while condensation is taking place in the shell condensers in the regular way, it is only necessary to start the pump on the liquid jet to turn the system into a jet condenser, and the condensers into liquid coolers. This makes it easily possible to determine under what conditions the jet of liquid is of added value or a detriment to condensation. The gas flowing to the condenser under testing conditions averaged about 175 degrees and the thermometers indicated that with a small amount of liquid spray the superheat disappeared instantly, or at least in the 18 inch distance between thermometers. When the amount of liquid was increased to the maximum, the only result in every case was to raise the condenser pressure from 5 to 10 lbs. in addition to the pressure shown when condensing by the normal method, without the liquid jet. This experiment was repeated a great number of times and always with the same result, which result while apparently paradoxical was found after careful examination to follow natural laws and consequently was logical.

We based the design of this jet condenser upon pure theory, in so far as we could anticipate, and we expected that it was going to revolutionize ammonia condensation, but like a great many other revolutionary devices in the refrigerating field it did not revolutionize and that is the reason that we have not told you of this before. The introduction of the liquid into a condensing gas under certain conditions retards rather than facilitates condensation and consequently raises the condenser pressure. In my opinion such conditions are present in the flooded type of condenser under discussion and this accounts for the contention of Mr. Torrance, that it is more efficient without the liquid jet than with it.

*Mr. Torrance.**—It must be remembered that the above mentioned tests were made consecutively on the same apparatus. If air were present it would affect both sets of experiments equally. Furthermore, after switching the valves, so that the condenser was operated on the injector method, the head pressure never dropped, hence the results obtained appear conclusive.

The writer believes the theory of condensation mentioned in the paper to be correct, that is, a "flooded" brine cooler is correct in practice and theory, but a "flooded" condenser is incorrect in practice and theory.

Mr. Greene's theory regarding the heat transmission is interesting, but we must consider it merely as a theory. The calculation as to the velocity of the vapor through the empty coil, Table II, is doubtless correct, but to prove its application to the flooded condenser we should build a condenser of glass, so as to observe and measure the shape and quantity of liquid that is drawn, suspended, dragged, or somehow maintained in the various parts of the coil.

It has been claimed that the flooded condenser is the greatest improvement made in the refrigerating industry in recent years, but these tests and those made a year ago indicate that this is not true. Now let someone else make comparative tests on similar apparatus, and, if they think they can prove that "flooding" is a benefit, then let this Society appoint a committee to investigate and report. Even if the cost is considerable, it will be worth while to fulfil our obligation to the industry.

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* Author's closure under the rules.

REVIEW

EXPERIMENTS ON THE RATE OF HEAT TRANSFER FROM A HOT GAS TO A COOLER METALLIC SURFACE. Babcock & Wilcox Company, New York, N. Y.

The experiments described in this book are of exceptional interest to refrigerating engineers, not only on account of the final results obtained, but also on account of the great care observed in eliminating the causes of error in the observations made and results calculated from them and the manner in which the theory of heat transfer as at present known is used to analyze the results obtained.

The phenomena of heat transfer are of great importance in the practical applications of mechanical refrigeration. The various papers presented before the Society in the last few years dealing with this subject show that reliable data either from a theoretical aspect or from a practical point of view as expressed by empirical formulae determined from actual experiments is not available. This lack of knowledge must have retarded to some extent the development towards greater efficiency of the apparatus used for the transfer of heat.

When conducting experiments to determine the rate of heat transfer between two fluids it is necessary to determine first which one of the fluids, when varying the quantity used within the range of good practice, will affect the rate of heat transfer to the greatest degree. In the case of properly designed steam boilers the circulation of the water evaporating is such that, as determined by the Babcock & Wilcox Co., even a considerable variation in the quantity of water circulated will not affect the temperature of the metallic heat transmitting surfaces sufficiently to have a marked effect on the rate of heat transfer of the boiler as a whole. The experiments made by them and described under the above heading for this reason are based on the rate of heat transfer from the hot gases to the cooler metallic surfaces only.

Even in the case of boilers a construction may be devised which will bring the hot gases into such intimate contact with the boiler surfaces that these surfaces will have nearly the temperature of the hot gases. A practical example of such a design is the flash boiler used to some extent in the automobile industry. The steaming quality of a flash boiler does not alone depend upon the rate with which heat is transferred from the hot gases to the metallic surfaces but also on the facility with which the water and steam can absorb this heat.

That the laws governing the rate of heat transfer in the heat absorbing apparatus most commonly used in the refrigerating industry are not of the same character as those pertaining to the operation of water tube or of fire tube boilers can be readily seen by the study of the flow of the brine and the ammonia in a typical can ice making tank equipped with vertical brine cooling coils. If each of the coils is fed by a separate expansion valve the ammonia will enter the bottom pipe partly as liquid and partly as vapor. As the ammonia proceeds up through the coils more and more of the liquid will evaporate until at the outlet ends only ammonia vapor issues. On account of the preponderance of liquid ammonia in the lower pipes of the coils it is fair to assume that the metallic surfaces in that part of the coils will be very close to the temperature of the liquid ammonia. The quantity of brine coming in contact with these surfaces would therefore affect the rate of heat transfer more than would an increase in the weight of ammonia circulated. This statement is borne out by the facts as gathered from existing plants. Whereas most can ice making tanks formerly installed were designed to produce their guaranteed capacity with a mean temperature difference of 15° and sometimes more degrees F. difference between the average brine temperature in the tank and the average boiling point of the ammonia in the coils, best practice to-day requires that this difference does not exceed 5° F.; even 3° F. can be readily and economically obtained. This change in the mean temperature difference required is entirely obtained by increasing the quantity of the brine that comes in contact with the lower pipes of the cooling coils.

For the upper pipes, in which the ammonia vapor preponderates, an increase in the quantity of ammonia passing the cooling surfaces is again of greater importance than an increase in the quantity of brine.

It is the lack of importance placed on a thorough knowledge of the factors that determine an increase in the rate of heat transfer, which has been the cause of the many failures to obtain the guaranteed ice output where standard brine coil equipments with separate expansion valves for each coil were changed to the so-called "flooded type" with one or two expansion valves for the whole tank.

The above may be briefly summarized as follows:

The rate of heat transfer between two fluids separated by a metallic partition and exchanging heat can be increased by bringing in contact with the heat transmitting surfaces a greater weight of the fluid whose average temperature differs most from the temperature of the metallic surfaces of the partition separating them. The relation of the two fluids to the rate of heat transfer may change in any particular apparatus in case either one or both of the fluids are liquids being evaporated or vapors being condensed, or it may be that, in some part of the apparatus, they may, through partial change of state or otherwise, by changing the weights circulated in any given time, together affect the rate of heat transfer.

The purpose of the tests made by the Babcock & Wilcox Co., as stated in the introduction, is to arrive at some definite conclusion on the laws governing heat transfer. The experiments of H. P. Jordan were in their estimation the most reliable of which a record existed at that time, but the method used by him in obtaining some of the data used to arrive at his results are qusetioned. The empirical formula used by both for expressing the rate of heat transfer is the following:

$$R = a + b\frac{W}{A} \qquad (1)$$

where R = the rate of heat transfer per square foot per hour per 1° F. mean temperature difference.

W = the weight of gas passing through the absorbing surface per hour.

A = area of the section of the channel through which the gas passes, in square feet, a and b were determined by experiment.

From carefully conducted boiler tests the Babcock & Wilcox Co. found for the average transfer rate with their boilers

a in (1) $= 2.0$ and $b = 0.0014$, while Jordan from his experiments found $a = 5.4$ and $b = 0.000506 - 0.00045r + 0.00000083\ (T + t)$, where r = the area of channel divided by the circumference in inches.

T = the mean temperature of the gas, °F.

t = temperature difference of the absorbing surface.

The main objection raised by the Babcock & Wilcox Co. to the experiments made by Jordan was the method he used for finding at any point of the experimental apparatus the mean temperature of the gas. Jordan led part of the gas away from the apparatus at various points to determine its average temperature, specific heat, etc. This method is objected to on the ground that by so doing the flow of the gas through the apparatus is disturbed, causing a condtion in the apparatus that would not exist if all the gas were allowed to flow from one end of the apparatus to the other.

The method used by the Babcock & Wilcox Co. was to determine the heat given up by the gas to the surrounding water and from this arrive at the average temperature of the gas at successive points of the apparatus, starting with a known measured average temperature of the gas where it leaves the transfer tube and determining its analysis, the weight flowing through the apparatus per hour and its average specific heat at the various points of the apparatus where the measurements were taken. The knowledge of the specific heats of nitrogen, carbon dioxide and water vapor as contained experimentally by Holborn and Henning were taken as the basis of these calculations.

The description of the experimental apparatus is given in great detail. It is sufficient to say here that the heat transfer tube used was a 20-foot copper pipe of 2-inch internal diameter surrounded by twenty individual water jackets each about one foot long. An illuminating gas furnace supplied the heated gases. The gases were drawn through the tube by an exhauster. After passing through the transfer tube, the gases passed through a cooler, then a dewpoint box, an equalizing chamber and then to the exhauster.

The dewpoint box was used to determine the density of the water vapor in the gases. This, together with the analyses of the gases, and the temperature of the gases entering and leaving the cooler, the mean sepcific heat of the gases through the cooler, can be calculated. This, taken together with the heat absorbed by the water passing through the cooler, made it possible to arrive at the weight of gases passing through the apparatus in a given time. Knowing the weight of the gases, the heat absorbed in jacket No. 20, the last jacket, and the temperature of the gases leaving this jacket, and taking the instantaneous specific heat of the gases from jacket No. 20 as the mean specific heat of the gases in jacket No. 20, the temperature of the gases entering jacket No. 20 and leaving No. 19 was calculated. This method was used in providing the temperature of the gases at the entrance to and exits from each of the 20 jackets.

The temperature of the metal surfaces of the transfer tube where measured by means of thermo-electric couples in the center of some of the jackets.

The heated water supply to the jackets was driven from a large cylindrical tank suspended above the transfer tube. This apparatus was so constructed that it was possible to furnish each jacket with the proper quantity of water, and at a uniform temperature, not to vary more than one degree to any of the jackets. The quantity of water used in each jacket was varied so that the outlet water temperatures from all the jackets was practically the same To prevent heat passing from one adjacent jacket to the other the water inlets to jackets Nos. 1 and 2 were placed adjacent to each other; the water outlets of jackets Nos. 2 and 3 were placed adjacent to each other and so on.

The jacket water was weighed through 6-minute intervals, which was also the length of each test.

The equalizing chamber was installed between the dewpoint box and the exhauster, to prevent fluctuations in pressure caused by the exhauster from effecting the flow of the gas in the transfer tube.

Preliminary experiments, covering a period of over three months, were conducted with a view of obtaining a full knowledge of the apparatus.

On test days the apparatus was operated the whole day, the morning period being used to adjust the apparatus to bring about steady conditions.

The data used for the final computations was taken in the afternoon and each reading was taken independently by two observers as a check.

To obtain the radiation losses from the apparatus the transfer tube was sealed and the temperature loss in the jacket water observed, as well as the quantity flowing through them.

The metal temperatures were taken by thermo-electric couples, carefully calibrated. A complete account of this work is given in every detail. The final formula deduced for determining the metal temperature at the ends of each jacket was as follows:

$$t_1 = T_1 + \Delta T = T_1 + \frac{H_1}{49 + 0.253\,W} \qquad (5)$$

where t_1 = the metal temperature at the end of the jacket.
T_1 = inlet temperature of water to a jacket,
ΔT_1 = difference in temperature between metal and circulating water,
H_1 = the B.t.u. absorbed per jacket per hour, corrected for radiation.
W = weight of water circulated through the jacket in pounds per hour.

Careful analyses was made of the computations to detect and correct errors and to obtain checks on recorded data and computed results. The method used is given in detail.

On account of the manner in which the tests were conducted it was possible to check by means of curves the heats absorbed in each jacket during any one experiment. It was found that when plotting the number of jackets against the logarithms of the heat absorbed in the jackets a straight line curve resulted, from which if any point varied a proper correction could be made using the following formula:

$$\log_{10} H_x = a + bx + cx^2 \qquad (6)$$

where H_x = the heat absorbed in one jacket per hour without radiation correction.
x = the number of jackets beginning at the furnace end
a, b and c are constants to be determined by the method of least squares.

To arrive at the transfer rate approximately for use in connection with the final formula the following equation was used:

$$R = \frac{H}{(T_m - t_m)\,S} \qquad (7)$$

where R = the transfer rate,
H = the heat absorbed.
T_m = the mean temperature of the gases.
t_m = the mean temperature of the metal.
S = the surface of the jacket in square feet.

The value so obtained was used in connection with the second part of the final equation for the transfer rate:

$$R = \frac{w\,c}{S} \log_e \left\{ \frac{T_1 - t_1 - (t_1 - t_2)\dfrac{wc}{RS}}{T_2 - t_2 - (t_1 - t_2)\dfrac{wc}{RS}} \right\} \qquad (8)$$

where R = the heat transfer rate in B.t.u. per hour per one degree difference in temperature per square foot of heating surface.
t_1 = temperature of metal at hot end of jacket.
t_2 = temperature of metal at cold end of jacket.
T_1 = gas temperature at hot end of jacket.
T_2 = gas temperature at cold end of jacket.
w = weight of gases flowing through pipe in pounds per hour.
c = specific heat of gases.
S = surface of jacket in square feet.

The value of R so obtained was again substituted in the equation and this operation continued until the value of R did not change for three decimal places.

A detailed account is then given with examples of the manner in which the gas weights were determined.

Finally the weights of gases per square foot of flue area per hour for constant temperature differences were plotted against the heat transfer rate and a family of curves obtained, all of which very closely approximate straight lines of the general equation:

$$R = a + b\frac{W}{A} \qquad (1)$$

Supplementary experiments were made by inserting a 1-inch outside diameter pipe inside of the 2-inch transfer tube. The results of these tests can also be approximated by formula (1), but the transfer rates is greater for the same gas weight and temperature difference.

In the discussion of the results it is pointed out that the points for plotting the final curve were taken from tests made on different days, under unavoidably different conditions and still show the desired uniformity.

It is further apparent from the final curves of heat transfer and rates of gas flow for constant temperature difference that, while to a very close approximation, equation (1) holds, as a matter of fact, the line represented by this equation is a secant or a tangent to a curve which bends from the line in both directions. The additional conclusion arrived at is that in case the actual design of a boiler varies to a great extent from the construction adopted for the experimental plant, as far as shape of heat transmitting surface, gas passage, etc., is concerned the constant a in formula (1) would vary considerably. This is borne out by the supplementary experiments with the core tube, where a is approximately equal to 2.5, while for the transfer tube without the core a is equal to 2.1.

To anyone interested in the subject of heat transfer and to those who wish to conduct experiments in connection with this subject, the material in this book is of exceptional value, particularly if carefully studied in connection with the chapter on heat transfer in the 35th edition of "Steam" hereinbefore referred to. It is hoped that this work will encourage other manufacturers of heat transfer apparatus to submit their designs to as close a study as the Babcock & Wilcox Co. have found it necessary and profitable to do in connection with the boilers manufactured by them. The material published forms a valuable addition to the existing engineering data.—*Fred Ophuls, Reviewer.*

HANDLING AND SHIPPING OF CITRUS FRUITS

HANDLING AND SHIPPING CITRUS FRUITS IN THE GULF STATES. By. H. J. Ramsey. Farmer's Bulletin 696, November 10, 1915. United States Department of Agriculture, Washington, D. C. 28pp. 10 figs.

In this bulletin the necessity for very careful handling of oranges and other citrus fruit is emphasized, this being absolutely essential in the financially successful conduct of the business of fruit growing. It is not sufficient to grow attractive fruit, it must be so harvested and handled as to minimize the losses from decay in transportation and at the markets. Only sound fruit brings the highest price.

The causes of decay in transit and after delivery are explained. Ninety-five per cent of the decay in fruit is caused by mechanical injury which can largely be prevented by the grower and shipper. In order to show what constitutes careful handling, the various harvesting operations are discussed separately, and in great detail. It is stated that co-operative and marketing organizations to secure the best results have found it advisable to take over and assume full responsibility for the field handling of their growers, employing trained crews to do the picking.

The packing house is usually located near a railroad, to minimize handling. The fruit should not be subjected to any drops or jars, and kept on one level as much as possible. Belt conveyors for the fruit boxes are extensively used. Two story buildings are to be preferred.

In many sections it appears to be necessary to wash and clean the fruit prior to packing to remove the sooty-mold fungus which follows the attacks of the white fly and other insects. Washing the fruit in water and drying it by the means available are the most difficult operations the fruit has to undergo, because too long soaking or insufficient drying under unfavorable climatic conditions is conducive to the growth of germs which soon spoil the fruit. In drying the fruit is exposed to a current of air, ordinary or heated. In this work there seems to be room for improvement. Drying by means of refrigerated (nearly moisture-free) air does not appear to have been tried.

Considerable space is given over to the questions of grading, sizing and packing. Boxes measure outside about 12½ by 12½ by 27 inches, and contain a space of 2 cubic feet, filled with 80 oranges of 4⅛ inch diameter to 324 fruit 2 inches diameter; average 176 fruit of 2⅞ inch diameter Boxes are usually loaded on end two tiers high into cars for shipment. Common loads contain 300, 330 or 360 boxes per car. Curing or holding the fruit a few days prior to packing with the expectation of throwing out fruit predisposed to decay, is not justified by the results. All fruit except lemons should be shipped as soon as possible after removal from the tree.

REFRIGERATION PRECOOLING

Up to within a few years practically none of the citrus fruit shipments from Florida were carried under refrigeration, because most of the crop is moved during the winter months. To guard against freezing the fruit is now transported in refrigerator cars with ventilation or refrigeration.

During the warmer periods and when decay is heavy, iced refrigerator cars are used, the cost of this icing being more than offset by the better condition of the fruit upon arrival at destination. It is pointed out that car icing does not offset the effects of rough handling, and that with sufficiently careful methods fruit can be satisfactorily delivered merely under ventilation. Besides in iced cars it takes five to seven days to lower the temperature of the fruit. During this time the germination and growth of blue-mold under very favorable moisture conditions occurs and largely nullifies the effect of this sort of refrigeration.

Precooling prior to shipment appears to be much more advantageous than ice refrigeration during transit. In this way the fruit can be promptly cooled down to about 40 degrees Fah. and decay largely arrested. In the Gulf States precooling was introduced only in the season of 1913–14. During two seasons the Department of Agriculture has conducted extensive and careful investigations of the advantages resulting from precooling of Florida oranges, and found it to be a valuable aid to careful handling in insuring sound condition of fruit on arrival and for a period of about ten days thereafter. In fact, precooling makes it possible to place on the market sound oranges throughout the shipping season, and requiring during the greater part of the season initial icing only, and during about three months the precooling alone is sufficient to insure safe transit. The combined cost of precooling and initial icing is declared to be considerably below the cost of full refrigeration, that is, initial plus periodical reicing.

Of the two methods of precooling, the car and the warehouse method, the warehouse method is more economical, because in this the loss of refrigeration is less, and the fruit is subjected to cooling sooner than in the case of the car which must wait until loaded and moved to the precooling plant. For small installations and where ice can be had at low cost, the ice and salt method for producing chilled air is advised, otherwise the more expensive ammonia refrigerating plants must be used.

No oranges or grapefruit are held in cold storage in the producing districts, but at the market end this is frequently done. Oranges should be stored at 38 to 40 degrees; lemons at about 42 degrees, and grapefruit at 45 to 50 degrees Fah. Oranges take on a brown stain in storage, and besides are just as susceptible to serious spotting due to attacks of Colletotrichum, a disease, as are grapefruit and lemons, especially at temperatures down to 32 degrees. Accordingly successful storage is limited to 4 to 6 weeks, or 9 weeks at the best. Frequent inspection is necessary to remove fruit showing signs of spotting.

THAWING FROZEN PIPES

PORTABLE EQUIPMENT FOR THAWING FROZEN PIPES. *Electrical World*, Vol. 67, No. 12, March 18, 1916.

A gasoline engine directly connected to a low-voltage direct-current generator, a switchboard, and a reel of cable assembled on a small truck make up an outfit that is being effectively used by the water-works department of the city of St. Paul, Minn., to thaw out frozen water pipes. The generator is rated at 20 kilowatts at 40 volts, and is driven by a four-cylinder, four-stroke-cycle gasoline engine. A voltmeter, an ammeter, a rheostat, and a single-pole knife switch of 500 ampere rating are mounted on the switchboard. The reel holds 500 feet of flexible copper cable of 300,000-circ. mil. cross-section in 100-foot lengths. Each length is fitted with terminal lugs. The various parts are attached to a steel frame, which is mounted on a 1.5-ton trailer hauled by a motor truck. The mounting is temporary, so that after the winter season the thawing unit may be removed and the trailer used for other purposes.

In operation, the positive terminal of the generator is connected through a section of the cable to the frozen pipe in the building between the meter and the street main and the negative terminal to the nearest outside fire hydrant, so that the circuit is through the lead or iron service pipe and the main to the hydrant. After the cables are connected, the engine is started and the voltage built up to between 30 volts and 40 volts, when the main swtich is closed. A current flow of about 250 amperes is maintained for two or three minutes and then increased to 350–500 amperes, depending upon how badly the pipe is frozen. Under these conditions it is reported that the service line has been cleared in all cases within ten minutes.

ECONOMY OF ICE PLANTS OPERATED FROM CENTRAL STATION

ECONOMY OF ICE PLANTS CONNECTED WITH ELECTRIC CENTRAL STATIONS. Eis und Kaelte-Industrie, Vienna, Vol. 8, No 9, March, 1916. 1½ pp. Review of article by Rich. Pabst, Cologne, in Elektrotechnische Zeitschrift, Jan. 20, 1916

Following the example of central power stations in the United States, German electric power companies are encouraging the supply of current to ice factories. During the past few years practical experience has been gained, notably from the 120 ton plant at Steglitz, Berlin, and from the 60 ton plant at Oberhausen, Rhineland, fully described in *Zeitschrift f. d. gesamte Kaelte-Industrie*, Munich, July and August, 1915.

Table I is submitted to show the estimated cost of operation in different size electrically driven ice plants. These plants are calculated to be operated for total 4,500 of the 8,760 hours of a year, equivalent to a load factor of 51.4 per cent.

The power consumption stated in Table I covers the complete installation, exclusive of cooling water supply, and is based upon using water with an initial temperature of 50 degrees F. With each degree F. above 50 degrees F. there is an increase of 2.2 per cent in the consumption of current and cooling water.[1] The prices given include all accessories, freight for 62 miles, and cost of erection. The ice plant is assumed to take current during 4,500 hours only of the year. At Steglitz and at Oberhausen the rate charged is 1.44 cent per k. w. hour. The cost of city water for ice making may be taken at 9.08 cents per 1,000 gallons, and that of well water for condensing at 2.73 cents per 1,000 gallons. For thawing the ice out of cans about 2 kg. or 4.4 llbs. of steam are required, which at $3.05 per ton f.o.b. central station amounts to a cost of 32.7 cents per 1,000 lbs. steam.

According to the author's calculation the manufacturing cost per 2,000 lbs.

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[1] With water at 105 degrees F. the consumption would be doubled.—Ed.

of ice in a 26½ ton ice plant is $1.65. In larger plants this cost diminishes to $1.30, while in smaller plants the cost will exceed $1.65.

Central stations manufacturing ice themselves usually contract with some one for the delivery of the ice. The ice is then sold at $1.96 per ton of 2,000 lbs. f.o.b. factory. Hence the profit to the ice maker is $1.96 – $1.65 = 31 cents per ton. In this case the return on the capital invested is 11.3 per cent.

But the central station profits further by the supply of current to the ice making department, because its cost of generating a k.w. hour is only 0.84 cent. By Table I the yearly consumption of current is 220,500 k.w. hours at a price of 1.44 cents. The gain is therefore 220,500 × $0.006 = $1,323, making the total profit on the investment 11.3 + 5.5. = 16.3 per cent.

TABLE 1.

Cost of Operating German Electrically Driven Ice Plants. Load Factor 51.4%

Output of ice in tons of 2,000 lbs. per 24 hours....................	6.61	9.92	13.22	19.83	26.45	39.67	52.90
Refrigerating machine capacity required, Calories per hour........	35,000	50,000	60,000	90,000	120,000	180,000	240,000
Tons of Refrig. per 24 hrs.........	11.57	16.52	19.83	29.75	39.68	59.5	79.35
Tons of Refrig. per 24 hrs. per ton of ice...........................	1.75	1.67	1.50	1.50	1.50	1.50	1.498
Total brake h. p. required (English)	18.75	24.67	30.6	40.4	54.2	74.	93.6
Total k. w. hours required.........	432	552	672	888	1176	1560	1968
Total k. w. hours required per ton of ice...........................	65.4	55.7	50.8	44.8	44.5	39.3	37.2
Total k. w. hours required per year	81,000	103,500	126,000	166,500	220,500	292,500	369,000
Consumption of cooling water in U. S. gallons per hour with submerged condenser...............	1,374	1,982	2,510	3,700	4,890	7,400	9,780
Consumption of cooling water in U.S. gallons per hour with atmospheric evaporative condenser...........	264	330	396	528	660	1,057	1,321
Approximate first cost of complete ice plant, incl. tank and pipe insulation (1 mark = 24 cents)....	$6,720	$7,920	$9,125	$12,240	$14,400	$19,680	$24,000
Approximate first cost of electrical equipment....................	$1,089	$1,128	$1,392	$1,583	$1,848	$2,088	$2,594
Cost of supplies per year, incl. ammonia, oil, salt, sheet and rod packing......................	$96	$108	$120	$138	$156	$180	$204
Ground area covered by building, incl. ice storage room, sq. ft......	2,262	2,478	2,692	2,800	2,910	3,080	3,226
Approximate cost of foundations and building, incl. insulated ice storage room.........................	$5,400	$5,760	$6,000	$6,240	$6,480	$6,720	$7,200

SOCIETY AFFAIRS

NEW YORK CITY REGULATIONS FOR REFRIGERATING PLANTS

REGULATIONS FOR REFRIGERATING PLANTS AS ADOPTED TO COMPLY WITH CHAPTER 10, ARTICLE 18, OF THE CODE OF ORDINANCES OF THE CITY OF NEW YORK

EMERGENCY DEVICE

1. All refrigerating plants of 3 tons refrigerating capacity or over, using ammonia or ethyl chloride as a refrigerating agent, shall be equipped with a deivce or devices for discharging the refrigerating agent under pressure, in case of emergency, into sufficient water to absorb all discharged gas and carry it off to the sewer. The device or devices to be constructed to conform with that recommended by The American Society of Refrigerating Engineers, March 20, 1916, as shown on Fire Department drawing "A."

The Fire Department will have sole use of this device, and supply the necessary water.

2. No valves shall be located in the emergency discharge lines, excepting the control and check valves to the water and refrigerant mixer, but there may be one control valve on each emergency line located near connection to main high and low pressure lines for repair purposes only. Such additional valves shall be kept open at all times, and if closed will constitute a violation of the intention of these regulations.

3. The emergency valves and Standard Fire Department Siamese Connection shall be located on the public thoroughfare side of the building, either outside of building or in a vestibule having glass panel doors, which will provide easy access from the street, with valve at a height of not more than 5 feet above street or vestibule floor level. In no case shall the emergency valves be located over the exhaust outlet for ventilating the room containing the refrigerating plant.

In large refrigerating plants, at the discretion of the engineer in charge, there may be installed pressure gauges on the high and low emergency lines for determining changes in pressure while emergency lines are operating.

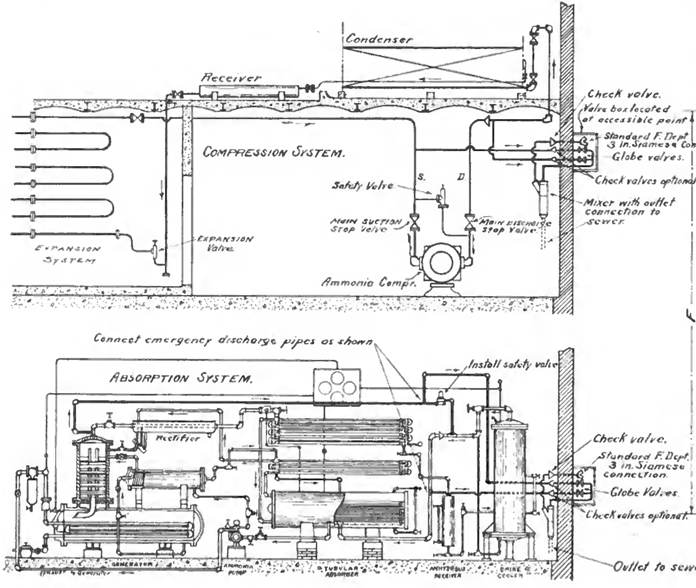
4. The emergency valves shall be protected by a suitable box, and kept locked. The lock shall be opened only by a "Fire Department Inner Box Key" and only by members of the Fire Department. On the outside of the box there shall be painted in easily legible letters the words "For Fire Department Use Only." The high pressure emergency valves shall be labelled "High Pressure Ammonia" and the low pressure emergency valves shall be labelled "Low Pressure Ammonia." A sign reading as follows shall be posted in the box: "Do Not Open Valves Until Water is Flowing. Maintain 50 lbs. Pressure."

AUTOMATIC SAFETY VALVES

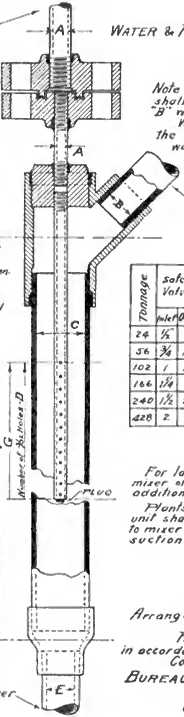
5. All refrigerating plants of 3 tons refrigerating capacity or over shall be equipped with an automatic safety valve or valves on each compressor or generator, which discharge at not greater than 300 lbs. pressure per square inch for ammonia, 1,400 lbs. pressure per square inch for carbon dioxide, 100 lbs. pressure per square inch for sulphur dioxide or ethyl chloride. Such automatic safety valve or valves shall be connected to the "High Pressure Side," and may discharge to the "Low Pressure Side," as recommended by

Drawing "A"

Proposed by the committee of the
American Society of Refrigerating Engineers.
March 20, 1916.



WATER & REFRIGERANT MIXER.



Note E:- The outlet from the ejector shall be four times the area of "B" wherever possible.

Where it is impossible to make the outlet four times the area of water inlet under standard conditions, a smaller size apparatus shall be used, adjusted to the possible size of the outlet.

Tonnage	Safety Valve		Diam of each emergency valve or pipe	inches			3/32 holes	inches		
	Inlet	Outlet		A	B	C	D	E	F	G
24	½"	1"	½"	1	1¼	5	266	See Note	50	15
56	¾	1¼	¾	1¼	2	6	460		60	20
102	1	1½	1	1½	2	7	625		70	25
166	1¼	2	1¼	2	2½	9	1000		90	30
240	1½	2½	1½	2	2½	9	1000		40	30
428	2	3	See note below				-		-	-

For large plants use one additional mixer of maximum capacity for each additional 240 tons or fraction thereof.

Plants containing more than one unit shall be independently connected to mixer or mixers from each main suction and discharge line.

Arrangement of Safety connections
for
Refrigerating Plants
in accordance with Chapter 10, Art. 18 of the
Code of Ordinances.
BUREAU of FIRE PREVENTION
Fire Department
City of New York.

The American Society of Refrigerating Engineers, March 20, 1916, and shown on Fire Department drawing "A."

The discharge from the automatic safety valve or valves may be conducted by continuous piping to a point at least 10 feet above the highest opening in the roof of the building in which the plant is located. Point of discharge above the roof to be located as remote as practicable from horizontal openings in surrounding buildings. Where ammonia is the refrigerating agent, the discharge end of the pipes shall be equipped with a diffuser, designed to mix the gas with the air wthout restricting its flow.

6. On carbon dioxide plants, in addition to the heretofore mentioned automatic safety valve at the compressor, there may be an automatic safety valve located on the high pressure and on the low pressure sides, as a substitute for the emergency device shown on the Fire Department drawing "A." The points of connection for such automatic safety valves shall be located as indicated for the emergency discharge lines. The outlet from such automatic safety valves may discharge into engine room or to outer air.

7. MAXIMUM PRESSURE ALLOWED

Refrigerating Agent		Pressure
Ammonia...........	not to exceed	300 lbs. per. sq. in.
Carbon Dioxide.....	" " "	1,400 " " "
Sulphur Dioxide.....	" " "	100 " " "
Ethyl Chloride......	" " "	100 " " "

8. All pipes used for refrigerating purposes shall stand a hydrostatic test of at least double the maximum pressure per square inch specified in section 7, and all fittings shall be guaranteed to stand a pressure of at least 3 times the maximum pressure per square inch specified in section 7. All gauge glasses shall be adequately protected and provided with shut off valves.

DEFINITION OF REFRIGERATING MACHINERY ROOMS

9. Refrigerating machinery rooms are rooms in which are located compressors, generators, condensers, receivers, absorbers or other apparatus necessary to produce or contain the refrigerating agent. Rooms which contain only the refrigerating chemical supply pipes to and from the refrigerating coils are not considered refrigerating machinery rooms, but such rooms, if occupied by any persons, must comply with the requirements as to exits hereinafter described for refrigerating machinery rooms.

LIGHTS, FLAMES, BOILERS

10. In all refrigerating machinery rooms in which ammonia or ethyl chloride is used, there shall be no flames, arc lights, gas jets or any apparatus employing flame, except internal combustion engines with internal ignition, which may be started in the usual manner.

11. All refrigerating machinery rooms shall be separated from other parts of the building by tight partitions and tight self-closing doors, and from boilers by tight fireproof walls unpierced, except by openings protected by tight self-closing fireproof doors or stationary fireproof windows.

EXIT AND VENTILATION

12. All refrigerating machinery rooms shall have a direct exit to the open

air or to a room or hall from which gases can be excluded by self-closing doors.

13. Vertical and horizontal openings that permit the passage of gases to other parts of the building, which are not a part of the refrigerating plant, shall be sealed or provided with self-closing doors.

14. All refrigerating machinery rooms shall be provided with adequate ventilation to the open air, either direct or by means of a suitable duct or ducts. Where a mechanical system of ventilation is employed, control of the system shall be located at a point easily accessible.

PLANS REQUIRED

15. Plans shall be filed for structural changes only. Alterations made necessary to meet the requirements of these regulations shall comply with the Building Code.

PIPES DESIGNATED

16. All pipes containing a refrigerating chemical under pressure shall have conspicuous signs attached to them at proper places designating in easily legible letters the name of the chemical contained therein.

The signs shall be placed on each main supply and suction pipe in each room. Valves which control main sections of refrigerating system, and are necessary to separate such sections in case of emergency, shall be labelled.

RULES

17. In all refrigerating plants there shall be posted several copies of a brief set of rules, satisfactory to the Fire Commissioner, directing all employees as to their duties in case of fire, or other emergencies.

Employers shall be responsible for the proper drill of all employees in such emergency duties.

HELMETS

18. All large refrigerating plants shall be provided with helmets of approved type, which will permit the wearer to reach, without suffocation, any part of the refrigerating system.

In plants of 25 to 75 tons refrigerating capacity there shall be at least one approved helmet, and in plants of greater capacity than 75 tons there shall be at least two approved helmets.

19. These helmets shall be kept in a suitable case or cabinet of easy access, preferably outside of refrigerating machinery room or near doorway leading therefrom.

SURPLUS REFRIGERANT STORAGE

20. Refrigerating plants may store a surplus stock of the refrigerating chemical of 2 cylinders, or, if necessary, a quantity not to exceed 10 per cent. of the charge of the plant. This surplus chemical shall be stored in a cool place.

CERTIFICATE OF FITNESS

21. No refrigerating plant of over 3 tons refrigerating capacity shall be operated unless under the charge of a person holding a Certificate of Fitness from the Fire Commissioner, or who has been duly certified as competent to operate a refrigerating plant by any Department of the City of New York having jurisdiction to certify thereto.

A Certificate of Fitness will be required of each person who is directly in charge of the operation of the refrigerating machines during each shift.

REFRIGERATING PLANTS OF LESS THAN 3 TONS CAPACITY

22. The Fire Commissioner may exempt from the provisions of these regulations, except the provisions referring to flames and exits, refrigerating machines of less than 3 tons capacity, provided a certificate of approval has been issued for such machine.

PERMITS

23. It is unlawful to operate within the City of New York any plant producing refrigeration by means of gases under pressure in connection with cold storage plants, breweries, ice manufactories, restaurants, or other places, without a permit.

The application for permit must state the refrigerating builders rated capacity in tons, the kind of chemical used, the amount of surplus chemical stored, the number of shifts and time the plant is operated during a period of 24 hours.

The application shall be signed by the applicant or authorized agent and give the name and address of the applicant and the address where plant is located.

COUNCIL NOTES

The Council held a meeting in the office of the Society, 154 Nassau Street, New York, N. Y., on the afternoon of March 24, 1916. The following officers and directors were present, with President Vilter presiding: Messrs. Vilter, Carpender, Borgstedt, Horne, Torrance, Broas, Matthews and Ross.

The Committee on Municipal and State Regulations for Refrigerating Plants and Refrigerants, in cooperation with officials of the Fire Department of New York City, submitted a tentative set of regulations and two drawings as to the manner of applying the New York City ordinance relating to refrigerating plants.* The Council felt that the check valves on the suction and discharge lines near the water and refrigerant mixer should be eliminated or their use made optional and the committee was requested to reconsider its suggestions.

The following letter from Charles C. Moore, President of the Panama-Pacific International Exposition was presented and ordered to be made a part of the records of the meeting.

THE PANAMA-PACIFIC INTERNATIONAL EXPOSITION
Office of the President

San Francisco, Cal., March 11, 1916.

Mr. W. H. Ross, Secretary,
The American Society of Refrigerating Engineers,
154 Nassau St., New York City.

My dear Mr. Ross:—

It was not possible for me to attend many of the convention sessions, neither was it possible for me personally to greet many of the delegates who attended the great number of congreeses, conferences and conventions which met with us during the Exposition period.

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* The completed regulations and diagram showing arrangement of safety connections are printed on another page of this number of The Journal.

I take this opportunity of expressing the appreciation of the Exposition management to the officers and members of your organization for the part they had in making the Exposition such an unqualified success. We consider the congress program one of the chief factors in producing that very gratifying result.

So many human interests were represented and the convention discussions had to do with such vital, practical, present day needs, that we are certain that true international fraternity and World service were distinctly advanced.

We know that convention attendance was largely responsible for the symmetrical success of our undertaking. We rejoice, however, that the Exposition was the means of making all of us better neighbors. You know us better. We are glad you came.

You will be interested in the enclosed "Final Word," which Director of Congresses, Jas. A. Barr, has prepared. It makes very clear why all of us who have been nearest the Exposition wish gratefully to acknowledge your loyal support.

Very truly yours,
CHARLES C. MOORE,
President.

The following report from N. H. Hiller, who represented the Society at the Second Pan-American Scientific Congress, was presented and also ordered to be made a part of the minutes of the meeting:

Carbondale, Pa., January 4, 1915.

MR. W. H. ROSS, Secretary,
The American Society of Refrigerating Engineers,
154 Nassau Street, New York City.

MY DEAR MR. ROSS:—

I am sending you, under separate cover, various papers and pamphlets that I received while attending the Second Pan-American Scientific Congress in Washington.

I do not know whether you desire a report on this Congress, but would give you the following facts, which may be of interest, viz:

Was unable to attend the Congress throughout, but was there on Tuesday, Wednesday and Thursday of last week. Dr. Dickerson, as the alternate delegate, attended the opening session and will take in the other social functions.

From a scientific standpoint, the business of the Congress was somewhat disappointing. There was but one paper, viz., that of Dr. Kent, that treated of thermotical subjects. All the others referred to civil engineering or electricity.

As the President of the National Geographical Society remarked, that for a Scientific Congress it was strange that one could find everything there but science.

The majority of the delegates from the South American countries were the attaches of the various embassies and were more interested in the political, than the scientific trend, of the work and that seemed to be the prevailing idea.

The sub-sections that I attended in the engineering section had to do with education, bridges and rivers and, though they were very interesting, were a little bit out of our line.

Taking it all in all, the proposition was rather disappointing and I can only account for it by the fact that it was handled by army engineers, whose only idea of engineering was something that had to do with water powers, bridges, waterways and electrical transmission.

I am very glad that there were no other members of the society there, as I am sure that they would not have felt that they had been repaid for the time and expense incurred.

Yours very truly,
N. H. HILLER.

The Membership Committee presented applications for membership from the following and they were approved in the grades indicated:

MEMBERS

Arnold W. Mindo, Melbourne, Victoria, Australia
O. Parke Seim, Seattle, Wash.
Edward D. Thurston, Jr., New York, N. Y.
J. Henry Beckman, Seattle, Wash.
Harry Sloan, Milwaukee, Wis.
F. W. Frerichs, St. Louis, Mo.
John N. Gruhl, Los Angeles, Cal.
William H. Schladitz, Cincinnati, O.
Harold E. Beavan, Redbank, Queensland, Australia

ASSOCIATE MEMBER

Godwin Werliin, Philadelphia, Pa.

JUNIOR MEMBERS

Emil H. Mack, Milwaukee, Wis.
Frederic T. Goes, Milwaukee, Wis.

CHANGE OF GRADE OF MEMBERSHIP FROM ASSOCIATE MEMBER TO MEMBER

Alan K. Gillespie, St. Louis, Mo.
George E. Chamberlin, St. Louis, Mo.

CHANGE OF GRADE OF MEMBERSHIP FROM JUNIOR MEMBER TO MEMBER

Donald Cole, Detroit, Mich.

The Council approved the further recommendation of the Membership Committee to invite W. H. Bower, Philadelphia, Pa., Hermann Dannenbaum, Philadelphia, Pa., and Louis Werliin, Philadelphia, Pa., to apply for change of grade of membership from Associate Member to Member.

The Council felt that the Year Book should be consoliated with the January number of the A. S. R. E. JOURNAL of each year, and referred this matter to the attention of the next annual meeting for final action.

A motion prevailed authorizing the printing of six hundred copies of the 1916 Year Book and the Treasurer was authorized to pay for them out of the funds of the Society.

The invitation of the National Fire Protection Association, inviting the Society to become a member of that association, was considered and The Council felt that the Society, through its committees, could thoroughly cooperate with the National Fire Protection Association without becoming a member of it.

A motion prevailed appropriating $600 for office rent for the year ending April 30, 1917. This sum to include increased space.

W. H. Ross was reappointed Secretary of the Society and two thousand dollars was appropriated to cover his salary for 1916.

An appropriation of two hnudred dollars was made for translations, editorial work, etc., on the A. S. R. E. JOURNAL.

An appropriation of six hundred and fifty dollars was made for the services of an office stenographer.

A special price of $1.50 for a yearly subscription to the A. S. R. E. JOURNAL to students was authorized.

The Secretary was authorized to insert an eighth page advertisement for three insertions in "Power," "Ice and Refrigeration" and "Refrigeration" advertising that non-members may secure a subscription to the A. S. R. E. JOURNAL.

The suggestion of John L. Porter, Chairman of the Cold Storage Section of the American Warehousemen's Association, that a joint meeting of the two organizations be held was considered and a motion prevailed requesting the Secretary to advise Mr. Porter that the dates of the annual meeting of the Society were selected so that they would immediately precede the dates of the meeting of The American Society of Mechanical Engineers, to which organization many of our members belong and it would be unwise to in any way change the dates, and the Secretary was further ordered to extend an invitation to the members of the American Warehousemen's Association to attend our meetings.

The resignations of the following members were accepted with regret:

L. C. Marshall, Boston, Mass.
John N. Briggs, Coeymans, N. Y.
W. W. Sandholt, Oakland, Cal.
Horace H. Judson, New York, N. Y.
Leonard W. Horr, La Grange, Ill.

Authorization was made to send copies of the A. S. R. E. JOURNAL regularly to the Scientific Station for Pure Products, New York, N. Y., and the Municipal Reference Library, New York, N. Y., in exchange for any publications they may issue.

The meeting then adjourned.

WILLIAM H. ROSS,
Secretary

TRADE CATALOGS

(Below is given a list of trade catalogs, etc., received at the Society's office since the last list was published in the A. S. R. E. JOURNAL.)

Crosby Steam Gauge & Valve Co., Boston, Mass.:
  General catalog.
  Crosby Spring Seat Globe and Angle Valves.
Vilter Manufacturing Co., Milwaukee, Wis.:
  High Efficiency Steam Engines, Bulletin No. 25.
Gifford-Wood Co., Hudson, N. Y.:
  Revised price list on ice tools, effective April 15, 1916.
C. H. Wheeler Manufacturing Co., Philadelphia, Pa.:
  Condensing Equipment.
Armstrong Cork & Insulation Co., Pittsburgh, Pa.:
  Nonpareil High Pressure Covering for Heated Surfaces.

STATEMENT OF THE OWNERSHIP, MANAGEMENT, CIRCULATION, ETC., required by the Act of Congress of August 24, 1912, of A. S. R. E. JOURNAL, published bi-monthly, at New York, N. Y., for April 1, 1916.

State of New York, County of New York, ss.:

Before me, a Commissioner of Deeds for the City of New York in and for the State and county aforesaid, personally appeared William H. Ross, who, having been duly sworn according to law, deposes and says that he is the Editor and Manager of the A. S. R. E. JOURNAL, and that the following is, to the best of his knowledge and belief, a true statement of the ownership and management of the aforesaid publication for the date shown in the above caption, required by the Act of August 24, 1912, embodied in Section 443, Postal Laws and Regulations, to wit:

1. That the names and addresses of the publisher, editor, managing editor, and business managers are:

Publisher—The American Society of Refrigerating Engineers, 154 Nassau street, New York, N. Y.

Editor—William H. Ross, 154 Nassau street, New York, N. Y.

Managing Editor—None.

Business Managers—William H. Ross, 154 Nassau street, New York, N. Y.

2. That the owners are: The American Society of Refrigerating Engineers, membership corporation organized under the laws of the State of New York and having no capital stock shares. There are 350 members. Officers of the Society are: President, Theodore O. Vilter, Milwaukee, Wis.; vice-presidents, Ezra Frick, Waynesboro, Pa.; and N. H. Hiller, Carbondale, Pa.; treasurer, George A. Horne, New York, N. Y.; secretary, William H. Ross, 154 Nassau street, New York, N. Y.

3. That the known bondholders, mortgagees, and other security holders owning or holding 1 per cent or more of total amount of bonds, mortgages, or other securities are: None.

4. That the two paragraphs next above, giving the names of the owners, stockholders, and security holders, if any, contain not only the list of stockholders and security holders as they appear upon the books of the company, but, also, in cases where the stockholder or security holder appears upon the books of the company as trustee or in any other fiduciary relation, the name of the person or corporation for whom such trustee is acting, is given; also that the said two paragraphs contain statements embracing affiant's full knowledge and belief as to the circumstances and conditions under which stockholders and security holders who do not appear upon the books of the company as trustees, hold stock and securities in a capacity other than that of a bona fide owner; and this affiant has no reason to believe that any other person, association, or corporation has any interest direct or indirect in the said stock, bonds or other securities than as so stated by him.

(Signed) WILLIAM H. ROSS, Business Manager.

Sworn to and subscribed before me this 22d day of March, 1916.

JOHN MCLAREN, Commissioner of Deeds.
(Seal) City of New York
(My commission expires June 23, 1916)

INDEXED

A. S. R. E. JOURNAL

INCLUDING THE TRANSACTIONS OF THE SOCIETY

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JULY, 1915

PUBLISHED BY
THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
New York, N.Y.

THE JOURNAL

OF

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

154 Nassau Street, New York, N. Y.

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A. S. R. E. JOURNAL

INCLUDING THE TRANSACTIONS OF THE SOCIETY

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PUBLISHED BY
THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
New York, N.Y.

A. S. R. E. JOURNAL

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NOVEMBER, 1915

PUBLISHED BY
THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
New York, N.Y.

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VOA

JANUARY, 1916

PUBLISHED BY
THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
New York, N.Y.

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A. S. R. E. JOURNAL

INCLUDING THE TRANSACTIONS OF THE SOCIETY

MARCH, 1916

PUBLISHED BY
The American Society of Refrigerating Engineers
New York, N.Y.

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JUN 9 1916

A. S. R. E. JOURNAL

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VOA

MAY, 1916

PUBLISHED BY
THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
New York, N.Y.

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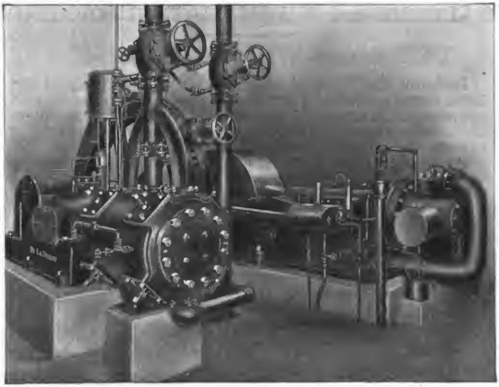

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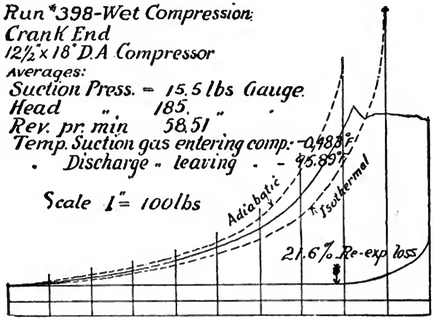
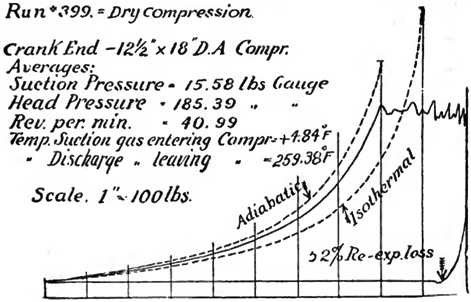
Indicator Cards are of very little help in arriving at the horse power per ton of a wet compression machine, for they do not indicate the amount of ammonia the compressor is handling, nor can the amount required to be handled be determined in any other way than by experiment.

Furthermore, indicator cards are of no use in determining volumetric efficiencies when taken from compressors operating under "WET CONDITION."

The following cards from the crank end supplement those shown in the March issue:





York Manufacturing Company

YORK, PENNSYLVANIA

DRY *versus* WET COMPRESSION

Referring to the cards, notice that the compression line on the wet compression cards is about midway between the isothermal and adiabatic curves, and that the loss due to re-expansion is about four times as much as is shown by the dry compression cards. Evidently the cause of the compression line swinging away from the isothermal curve is the gas formed by vaporization of the liquid, which kept the temperature down in the compressor.

The excessive re-expansion is due to the presence of liquid in the clearance space. The total loss from the vaporization on the suction stroke of the compressor is not shown by the cards, but this loss must be considerable.

In all of our "WET COMPRESSION" tests the amount of liquid allowed to come over to the compressor from the evaporating system, was the minimum amount that would keep the discharge temperature at the required point. This precaution, the result of careful observation and experiments, is not exercised in the ordinary plant, operating under "WET CONDITION"—consequently the efficiency of the compressor must necessarily fall below that tabulated above.

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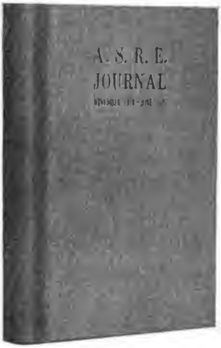

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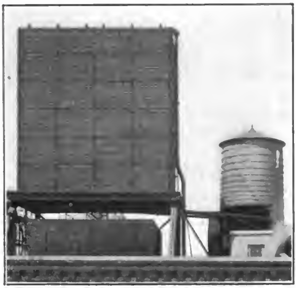

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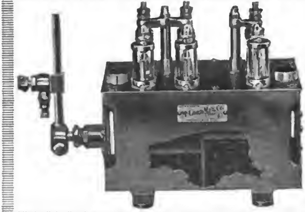

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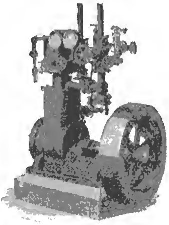

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